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CONTENTS

DECEMBER 1942

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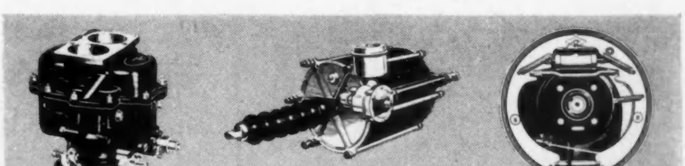

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HIGHWAY to VICTORY

by
WILLIAM J. CUMMING

Chief, Vehicle Maintenance Section, Division of Motor Transport, Office of Defense Transportation

TO make sure that we cross the finish line ahead of our Axis competitors, our Government has found it necessary to discontinue the production of civilian automotive vehicles for the duration. This means that highway transportation is to be cut off from the phenomenal progress that metallurgists, designers and production men have made during the past 30 or 40 yr because the efforts of these men must be devoted solely to the production of military materials. The maintenance engineer is on his own.

The baby of the automotive industry is the scientific operation and maintenance of the products the industry makes. The Transportation & Maintenance Activity of the SAE has been in existence since 1929. Since that time, we operators, as a group, have made progress. Sometimes it has been a little slow and awkward, but we have always gone forward.

Just before World War II, the T&M Activity embarked upon an ambitious and constructive program. It was an excellent peacetime program and in due course was bound to make itself felt as we reduced to a formula some of the uncertainties of fleet operation. Then came the War! It was necessary to redesign that program to fit the times and to supply the Office of Defense Transportation with the information it required. Joined with ODT in what John L. Rogers, Director of the ODT Motor Transport Division, has called a "war assignment," are over 250 T&M members and others interested in transportation and maintenance matters. All members are volunteers and are working diligently gathering information on thirty conservation subjects. All involved consider this effort a worthwhile patriotic duty. We could use as many more volunteers.

America's existing highway transportation plant *must* be properly maintained, and to that end Mr. Rogers set up a Vehicle Maintenance Section in the ODT Division of Motor Transport to develop a program to assist in this endeavor. Naturally, this program is one which the truck owner must voluntarily accept, as there is no way to enforce preventive maintenance at this time.

The first item of the program covers publicity releases on maintenance and repair procedures; on reclaiming, rebuilding and rehabilitation processes; on the urgency of the maintenance situation with regard to trucks and their mechanical units; on safety, mechanic and driver training;

[This paper was presented at the SAE War Transportation and Maintenance Meeting, New York, N. Y., Oct. 7, 1942.]

and on the need for older mechanics who have retired or sought less strenuous work to rejoin the active ranks of automotive maintenance as instructors or gang leaders. All types of releases pertaining to the operation and maintenance of automotive vehicles will be issued by the ODT Information Division, together with radio and movie presentations.

The second item covers the issuance of a maintenance booklet entitled "America's Trucks . . . Keep 'Em Rolling." This pamphlet contains an appeal to two and one-half million truck owners and repair men to do their part to prolong truck life and conserve gasoline, parts and tires. To do this, the truck must have special maintenance attention, and the booklet is a handy guide to a definite plan. At the present time four million copies have been printed and distributed.

■ Thirty Projects Under Way

Factors of maintenance control and procedure, another item, contemplate the issuance and follow-up – through established professional technical societies and associations – of suggestions relative to important maintenance techniques and procedures, such as standard maintenance instructions, unit replacement control, frequency of mechanical failures and reclaiming methods. It is felt that a great deal can be accomplished if a more thorough understanding of these comparatively simple fundamentals can be had by owners, as well as by the more than 100,000 repair agencies which exist in this country. At present thirty separate projects are being studied by the SAE Transportation & Maintenance Activity.

The matter of parts availability and supply, is another of the important functions of the ODT Maintenance Section. Thorough knowledge of the processes necessary to manufacture automotive parts of every type provides the basis for such mechanical procedures as may be set up to reclaim parts when worn. Conservation of worn material by processes known and practiced by the fleet operator for years is one of inestimable value at this time of shortages in important metals.

Building up worn areas by approved methods, rather than scrapping whole units because of wear at a time when there is little chance of getting new materials, is of definite value. ODT Maintenance Section personnel has the feeling that the word "salvage" in connection with this type of conservation does not properly describe such processes. Salvage hints of the scrap pile, when as a matter of fact most worn materials are far removed from the junk classifications.

A further explanation of this item of the ODT program contemplates investigations to determine: first, the availability of replacement parts at or in (a) parts manufacturers' supply stocks, (b) jobber-dealer stocks, (c) vehicle manufacturers' stocks (including branches and distributors); second, survey of parts needed by vehicle operators to determine their comparative need in value and volume in relation to present supply; third, information as to future potential need of replacement parts (based on inability to obtain new replacement vehicles); fourth, present and future sources must be analyzed to determine (a) Will present sources continue to produce in volume? (b) Are Government military orders reducing ability to produce parts? (c) Can present sources be enlarged? (d) Can inadequate sources be recreated? (e) Must new sources be created?

With these situations determined, it seems likely measures can be taken in cooperation with other Government bodies to anticipate the possibilities of shortages. In the past some of us have thought we knew quite a lot about maintaining fleets, but this war has really put us through school. We feel confident we can pass the course, most of which we will have to lay out for ourselves. By the time the war is over, we will have an interesting background and I believe our experience will have a profound effect upon the future of highway transportation. A new technology will have been born to parallel that of the metallurgists, designers and production men. When we pool our experience with theirs, I am sure that highway transportation will reach an efficiency never known.

These war conditions that often make it impossible for us to get replacements have thrown the spotlight on the maintenance man in dramatic manner. The drama is heightened by the lack of actors on the maintenance stage. There are just simply not enough good T&M men to go around and we believe our present SAE-ODT program is designed to do something about it. By the time the curtain call comes at the end of the war, there will be more.

Our SAE-T&M committee reports are the backbone of an ODT educational campaign. They will go a long way towards educating us in conserving our precious highway transportation equipment. Do not get the idea that a few of us are looking down our noses at the rest of the gang and condescending to teach them what we know. These committee reports are being used for educating all of us, including the committee members. Our approach has been based on the fact the automotive industry abounds with specialists and with fleet operators who know one phase of our job better than the rest of us because they have been interested in pioneering a single process or method.

We are taking advantage of that specialty or interest, by making available to all the findings of the few. Where one man is strong, another is weak, and we are working at the job of building up the weak spots of all of us to the strength of the specialist or pioneer. It is a never ending job because there will always be pioneers in mechanical processes, but by continuing, I believe, we can shorten the gap between the pioneer and the field.

■ Stretching Out Vehicle Miles

We can and are, as a result of our lack of replacements, operating trucks and buses many more miles than we or the designers thought had been built into them. In stretching out vehicle life, we are compiling a record of failure frequency that we could never get any other way. When the war is over and the industry goes back to building new equipment, we will be able to tell our designer friends what parts of the equipment failed and with what frequency. Since we have to keep the equipment rolling in face of parts shortages, we have found out some things about these failures and we expect to find out more. If our designer friends are interested, we will probably be able to tell them what causes the failures and what must be done to prevent them.

Of one thing you may be sure. After we have operated the vehicles for many more miles than we had been led to believe was in them, we will never be satisfied to go back to the shortlived vehicle. While we feel that *we* have been at fault, as well as the manufacturer of the vehicles, for short vehicle life, we are willing to admit it

and correct the condition so far as we are concerned. We are going to be tough to deal with if the manufacturer is not willing to meet us halfway.

Some of the corrections in vehicle design – or should we call it new design for changed conditions – will cost money. It is fair to ask where the money is coming from because I am not prepared to say that fleet operators as a class are going to be willing to pay more for vehicles in the future than they have in the past, even though future vehicles may have a longer life. I am afraid that too many salesmen who have tried to sell quality over price in the past would be able to argue the point all too well.

However, it does not seem imperative to increase the price because we think we have found where some money can be saved without impairing the operation of the vehicle in any measurable quantity. Since there has not been any material available to make these costly, heavy grilles, trim strips and other ornaments, we have been doing pretty well without them. Many of them have long since fallen off along the road and found their way to the junk pile where we hope the reclaimed material is doing more good for the armed services than the finished product ever did for us.

■ Will Be Better Buyers

After the war, truck operators will be better buyers of equipment than ever before, I believe. With frequency of failure reports before us, we will have a clearer idea of what it is we want to buy. This will make us better buyers, not only for ourselves but, I think, better buyers for the manufacturer. Our ideas will be pretty firm, and once we have made them understood by the vendors, I believe our relations will be even happier than they were before. We cannot, however, fail to pause and shed a tear for some types of salesmen who, good fellows though they were, made a fetish of failing to learn anything about highway transport. With nothing to sell at present, they have drifted away to other fields and all we can do is wish them happy hunting because, as we become better buyers, there is going to be less room for amateurs. For the salesman who knows the business and can bring us facts, there will always be a welcome.

To buy what we want, we will have to get into specifications. The minute we do that, we will have to get together with our friends in the designing rooms and show them why we have the specifications – and that brings us right back to a frequency of failure record and to war-born experience in operating vehicles longer than they were intended to operate.

Let me make it clear that fleet operators do not intend to install drafting rooms or laboratories and go into competition with manufacturers. Such specifications as we have developed in the past and will in the future are largely corrective specifications based upon our frequency of failure records; performance specifications based upon our past experience; and load specifications which simply modify existing design to tailor it to our individual containers or loading requirements. We have no desire to tell manufacturers how to build vehicles.

I am deeply impressed with the functioning of the various SAE-U. S. Army committees in which the SAE membership of the committees is composed of the proper manufacturing representatives for the problem at hand and the Army representation includes fleet operators faced with the operating problem. These committees provide an excellent exhibition of cooperative solution of industry problems.

I suggest to the SAE membership at large that, when the war is over and we return to our peacetime pursuits of improving highway transportation, the men in uniforms be replaced by civilian operators who, despite the fact that they lack epaulets, have problems. If this suggestion is not feasible, let's find another way to do the job because I am certain that fleet operators will never again fall for the story that if we buy model XYZ at so many dollars f.o.b., all of our troubles will disappear.

Already wartime experience has taught us enough about one topic to say that all subsequent discussions will vary only in the relative strength of the adjectives used. I refer to accessibility. With the man hours lost as a result of foolish locations of adjustments and the can opener technique required to remove some units alone we could win a full sized war. When we refuse to go back to the short-lived vehicles we will also refuse to go back to vehicles that require mechanics to be contortionists – with patience – if for no other reason than that the mechanics will no longer work under those conditions.

Inaccessibility has had a serious effect on our business, but we do not yet realize it. I was deeply shocked to learn recently that one of our best and largest trade schools had closed up its automotive courses. The reason? No lads were interested in becoming automotive mechanics.

We are willing to admit that, as a whole, our shops have not been models of cleanliness – nor has the automotive mechanics' rate of pay been all that it should be.

As we continue in the highway transport business, we are going to clean up the shops and we are going to pay the mechanic more money. When we pay the automobile mechanic more money, we are going to want him to produce more. This means the mechanical puzzles now presented by modern design will have to give way to something that takes less time to solve. No fleet operator is in any position to pay mechanics to try to outguess a designer who hides all available means of making adjustments and repairs.

■ Accessibility Cuts Maintenance Costs

To me, this utter disregard of location and accessibility of units that need maintenance attention is a reflection of the philosophy of the manufacturer. Apparently, two things have governed his thinking and maintenance. Accessibility is neither of them. One reason for inaccessibility is that manufacturers have worked on the basis that if they built the vehicle cheap enough, there would not be reason to complain about the hidden adjustment very often – and since it is cheaper to build without considering maintenance, the purchase price has been brought down somewhat at the expense of operating cost.

The other governing factor appears to have been sales sex-appeal. Operators do not want to pose as artists. We do want good-looking vehicles but not at the expense of accessibility. The truck designers who specialize in appearance seem to have stolen all of the shoddy features of passenger car appearance, forgetting all the while that after 40,000 miles the average passenger car is destined to find its way to a used car lot, while at the same mileage a truck is just becoming a serious headache to the maintenance crew.

turn to page 29

Sheppard photo

SAE War Engineering Board and guest engineers meet in the Rackham Memorial Building at Detroit every third Friday morning. At front table, left to right, are, Dr. J. S. Laird, Ralph N. DuBois, J. M. Crawford, R. R. Teetor, C. R. Paton, L. R. Buckendale, Louis Thoms, and F. F. Kishline. Rear table, left to right, Secretary R. C. Sackett, Don Blanchard, Lt.-Col. J. H. Frye, Chairman James C. Zeder, B. B. Bachman, John A. C. Warner, and H. H. Hughes

W.E.B. DETROIT!

From Cold-Proofing Motors Against Arctic's Chill to Heat-Proofing Electrical Apparatus for Tropical Service, Agenda of SAE's War Engineering Board run Gamut of Technology!

by VICTOR H. SCALES

AUTOMOTIVE engineering genius has been shifted by the SAE War Engineering Board from streamlining for civilian consumption to straightlining for the national war effort with success which suggests revision of that well-known slogan from f.o.b. to W.E.B. Detroit.

For the duration, the engineering miracles being worked by the Board largely must remain military secrets. Yet the enemy can derive no particular comfort from the knowledge that the War Engineering Board is concentrating the talent of top-flight aeronautical and automotive engineers upon a direct-action advisory service essential to the military requirements of a big nation engaged in mechanized warfare.

The work of the Board is directed by technologists who, experienced both in methods and materials, occupy executive positions which command the skill, data, knowledge, and experience of an immense cross-section of the nation's manufacturing capacity. Corporate identities and industrial competition are submerged for the duration. Engineering policies, research findings, laboratory and test equipment, accumulated experience, and trained personnel are available without reservation.

The Board has functioned since Pearl Harbor and now meets every three weeks. Its 16 members could display no greater devotion to duty were they in uniform. Their attendance record, which would be remarkable at any time, now is an earnest of the sincerely serious interest of these executive engineers from nine industrial cities who add Board work and travel to heavy corporate duties related to essential production for victory. With patriotic regularity they come from their offices in Racine and Kenosha, Wis.; Ardmore, Pa.; Paterson, N. J.; Indianapolis, South Bend, and Hagerstown, Ind.; Toledo, Ohio; and Peoria, Ill., as well as from Detroit's own huge manufacturing plants.

From the first the manpower problem of the Board has been one largely of selecting experts fitted by experience and ability for specific tasks. So far as availability and patriotic willingness to serve are concerned, Board membership easily could be 160, or 1600, or even 16,000. In fact, it is probable that close to 1600 outstanding engineers and technologists actually are engaged in conducting the various undertakings directed by the Board. The drop of an idea would be sufficient to bring the 14,400 others into service.

This volunteer effort, characterized as an outstanding example of close and effective cooperation between industry and the military, in a sense is the modern equivalent of an aroused citizenry springing to arms for the service of the country. It so happens that in this particular war

Blue-bound reports and recommendations of the SAE War Engineering Board are becoming a blue-book of wartime automotive engineering comprising 20 volumes — and still growing

engineers with slide rules are as necessary as soldiers with rifles; in patriotic spirit there is no difference.

Organization of the Board itself is a practical and substantial civilian contribution to the national war effort. It gives Government and military agencies the advantages of a concentration of brainpower which could not be hired, of administrative ability infrequently assembled, and of technological service not otherwise available. The costs, which by no compression of the imagination can be viewed as inconsiderable, are borne by industry. Machines and equipment, materials and supplies, laboratories and trained personnel, shops and testing grounds used in the Board's extensive research activities are loaned by industry without delay or question.

The Board's work means savings in cost to taxpayers, savings in time to the war effort, which are so tremendous as to be incalculable. Equally difficult to assess are the Board's ideal of selfless service, its purposeful desire to

Members of the SAE War Engineering Board

J. C. Zeder, Chairman
— *Chief Engineer, Chrysler Corp.*

B. B. Bachman
— *Vice President and Chief Engineer, Autocar Co.*

D. R. Berlin
— *Aeronautical Engineer, General Motors Corp.*

L. R. Buckendale
— *Vice President for Engineering, Timken-Detroit Axle Co.*

R. E. Cole
— *Vice President for Engineering, Studebaker Corp.*

J. M. Crawford
— *Chief Engineer, Chevrolet Div., General Motors Corp.*

F. F. Kishline
— *Chief Engineer, Nash Div., Nash-Kelvinator Corp.*

R. H. McCarroll
— *Ford Motor Co.*

Arthur Nutt
— *Vice President for Engineering, Wright Aeronautical Corp.*

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— *Executive Engineer, Allison Motors Div., General Motors Corp.*

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D. G. Roos
— *Vice President and Chief Engineer, Willys-Overland Motors, Inc.*

C. G. A. Rosen
— *Director of Research, Caterpillar Tractor Co.*

E. H. Smith
— *Executive Engineer, Aircraft Div., Packard Motor Co.*

R. R. Teetor
— *Vice President, Perfect Circle Co.*

Louis Thoms
— *Central Engineering Div., General Motors Corp.*

R. C. Sackett, Secretary

General Barnes says . . .

ALL COMMUNICATIONS SHOULD BE ACCOMPANIED BY CARBON COPY AND ADDRESSED TO

WAR DEPARTMENT
OFFICE OF THE CHIEF OF ORDNANCE
WASHINGTON

September 28, 1942

War Engineering Board
Society of Automotive Engineers
917 New Center Building
Detroit, Michigan

Gentlemen:

A review of the results of the conservation work of the Ordnance Department indicates that enormous quantities of critical or strategic materials have been saved. This is especially gratifying when it is considered that this conservation was accomplished without sacrifice of safety or military efficiency of the materiel.

The SAE-War Engineering Board has aided materially in accomplishing this result. The experience, engineering skill, and untiring efforts of the Board members in their study and subsequent recommendations for the materials used in the construction of military vehicles have proven of invaluable assistance.

It is a pleasure to commend the War Engineering Board and each of its members for this contribution to the war effort.

For the Chief of Ordnance:

SENT SEP 29 AM 42 ORDNANCE MAIL AND RECORD BRANCH WAR DEPT.

Very truly yours,

G. M. BARNES
Brig. General, Ord. Dept.
Chief, Technical Division

Under date of Sept. 28, 1942, Brig.-Gen. G. M. Barnes, chief of the U. S. Army Ordnance Department's Technical Division, wrote that through conservation work enormous quantities of critical or strategic materials had been saved without sacrifice of safety or military efficiency of the material, and added:

"The SAE War Engineering Board has aided materially in accomplishing this result. The experience, engineering skill, and untiring efforts of Board members in their study and subsequent recommendations for the materials used in the construction of military vehicles have proven of invaluable assistance.

"It is a pleasure to commend the War Engineering Board and each of its members for this contribution to the war effort."

achieve one objective only to seek another. To characterize the Board merely as a mobilization of engineering talent, however, is to be miserly of description. It is, in effect, a technological general staff continuously engaged in strategic master-minding upon the battlefields of design, construction, maintenance, and research engineering.

To make description even reasonably complete, it must be added that the Board has become a living example of the catalytic power of patriotism. Before the war the very idea of executive engineers from competing corporations in widely-scattered cities gathering at Detroit for discussion of mutual engineering problems, exchanging blueprints and reports, sharing research facilities and findings, trading ideas and experiences – once secrets as closely guarded as the nature of the bugs in next year's model – would have been beyond the pale even of contemplation. To have invited into a planning and research department an engineer from another plant for the open and obvious purpose of revealing laboratory developments would have been unthinkable.

These are matters of course today. The council table around which the War Engineering Board cogitates every third Friday in the Rackham Memorial Building is the altar of industrial sacrifice. There are substantial reasons to believe that table is becoming a technological plateau to which the engineering capabilities of the whole gamut of so-called automotive industries are being lifted through the mutual sharing of knowledge and which, after the war, will be used by competitors as the foundation of new and greater automotive achievements.

A meeting of the Board offers convincing proof that World War II will be no exception to the fact that however destructive of human happiness war may be, it is also an effective agent for accelerating human progress. The engineers who comprise the Board are too much concerned with the pressing present to devote talk or thought to post-war vehicles and planes, but conclusion is inescapable that their wartime cooperation and experiences will result in the design and production of land and air conveyances so advanced as to relegate pre-war models almost to the category of museum pieces.

Judged by war's more glamorous standards, the work of the Board is neither thrilling nor spectacular. Instead, it comprises those ditch-digging and foundation-laying, those pioneering and trouble-shooting jobs, which are the stuff of which victories are made, but normally are overlooked when the bands escort the conquering heroes up Main Street. Furthermore, the smoothly engineered efficiency with which the Board operates and the facility with which it mobilizes and directs engineering brainpower give its work a deceptive patina of minutiae which conceals the cold, hard fact that modern mechanized warfare must be won first in engineering laboratories and shops.

In 10 months this Board has undertaken, carrying out directives of Government and military agencies, to complete more than 30 fundamental projects in addition to the Board's vital work for solution of the critical national problem of tire shortage. These projects have called for extensive research and exhaustive tests, for the drafting of reports and the preparation of recommendations, upon which well may depend the successful operation of military equipment on and above fields of battle around the world, as well as the continued operation of civilian automotive equipment in essential services.

Thirty-six reports have been made to the Army and other Government agencies in recent months by the SAE War Engineering Board.

These reports, ranging in length from two to 62 pages, were developed upon specific request of the Government agencies, and required a total of tens of thousands of man hours in developing basic material, and hundreds of man hours of committee work on the part of the Board's members.

The Board's straightlined efficiency has reached the point where reports are ready for consideration and recommendations receive final approval within a period only of days. The reports and recommendations vary in length from three to more than 60 pages. They represent the work –

much of it pioneering effort – of hundreds of men in dozens of laboratories and shops. Not infrequently these documents incorporate technical data never before compiled.

The scope and the variety of the reports and recommendations indicate there is, apparently, no physical limit to the range of technological problems referred to the Board for solution. Upon receipt of a directive, which by the Board is regarded as another opportunity to get on with the business of winning the war, individual members volunteer, or are appointed, sponsors of the project created by the directive. Thereafter the sponsors are charged with the responsibility for carrying the project through to completion – and promptly.

In consultation with members of the Board, the sponsors select from industry the best qualified engineers and other experts to be chairmen and members of project committees and subcommittees. When, as rapidly as is consistent with sound engineering practice, the work of each project is completed, a covering report is submitted for the Board's consideration. Upon approval, this report becomes the basis for the Board's recommendations, which are forwarded to the appropriate Government or military agency. In the case of a project involving prolonged research or tests, one or more interim progress reports are made.

Since the Board functions only in an advisory capacity, further directives necessary to make the recommendations effective must come from the agency for which the work is completed. All reports and recommendations, from the initial data to the final, are held in confidence, supplied only to those who have received official approval.

■ Technological Ammunition

These succinct yet studied and complete reports, frequently involving expensive and extensive study and research, represent not only the contributions of industry to the national war effort, but, in effect, technological ammunition available to the war forces of no other nation. Yes and no answers are regarded as inadequate. Final recommendations are complete, supported by reasons, proofs, references, and the results of practical experience. In case experience is lacking, which is not unusual in view of rapidly changing conditions and also new fields of activity for American armed forces, the Board and its committees find ways and means of developing that experience or adequate equivalents.

Such inventiveness has been characteristic since the Board was organized in October, 1941, at the request of the SAE War Activity Council as the SAE Automotive Technical Advisory Committee on Critical Materials. Between Oct. 24, 1941, when Chairman James C. Zeder called to order the first meeting of the committee at Detroit, and February, 1942, automotive manufacturing activities shifted completely from civilian to military requirements. Accordingly the little group of technologists transferred its major interests from devising ways and means of meeting materials shortages to anticipating and solving technical problems imposed by wartime regulations, and to conducting an engineering advisory service for Government and the military.

With the change in objectives, serious straightlining set in. The group's name was trimmed to a title properly descriptive of its new work. The membership was increased. Outstanding technologists of the automotive, aircraft, and related industries were recruited for wartime service on a when, as, and if their talents were needed basis.

Whereas the group initially had worked with and for the Automotive, Transportation, Farm Equipment, and Conservation Branches of the Office of Production Management, the Office of Fuels Conservation, and others, it now became technical adviser to a complete new line-up of organizations. These include the War Production Board; the U. S. Army, its Ordnance Department, including the newly-organized Tank-Automotive Center, and Quartermaster Service; the U. S. Navy and its Bureau of Aeronautics, Bureau of Ships, Bureau of Ordnance, and Naval Gun Factories; the Automotive Council for War Production; the SAE War Activity Council; and others.

The number of agencies served bespeaks the wide variety and importance of the Board's engineering projects. During its pioneering days, many projects served helpfully to meet emergencies created by the industrial transition from partly civilian to completely military manufacturing operations. Current projects largely are related to military requirements; gradually are growing in number, in importance, and in difficulty; yet are confined exclusively to engineering.

Some of the problems referred to the Board for solution have come within the fields of experience and prior study; others have been problems of adaptation; still others have called for outright pioneering. Reduced, for purpose of understanding, to a statement so simple as to be insultingly deflating, the function of the Board is to engage both in

turn to page 25

Tires made of substitute materials but promising satisfactory performance are getting nearer realization and use as a result of the work of the Automobile and Rubber Industries Tire Committee. Reports and recommendations, many of them the basis for Government action, now comprise seven volumes

B. E. Sibley, chief technologist, Continental Oil Co., general chairman of the SAE National Fuels and Lubricants Meeting, Tulsa, at the right. C. M. Larson, chief consulting engineer, Sinclair Refining Co., and SAE Fuels & Lubricants vice-president, center; and C. W. Georgi, Quaker State Oil Refining Corp., one of the speakers, left

SAE Fuels and Lubricants Meeting Hears of War-Winning Petroleum

NEVER before in the active life of the Southwest Group of the Society of Automotive Engineers have its members and visitors learned so much of the vitally active work being carried on by the automotive and aeronautical forces of the Nation as they learned during the two-day SAE National Fuels and Lubricants Meeting at Tulsa, Oct. 22 and 23.

Servicing aircraft and maintaining planes, tanks, trucks, jeeps in the field; keeping oils in condition to speed up output and operation of military and allied automotive equipment; reducing down-time and on-the-ground time; reducing the number of men and machines needed on the ground to keep the fighting forces fighting; testing fuels; keeping engines working longer and better by preventing bearing failures; simplifying specifications to minimize the number of grades and types needed, and the fundamental cooperative research to make better conditions possible, were subjects discussed.

Herrington pleads for realism

With a thinly veiled condemnation of "little men" in Government, the armed services, in industry and labor, whose "personal grudges, petty vanities, abysmal greediness, and selfish aims" obstruct the proper conduct of the war and the defense of the democratic way of life, SAE National President A. W. Herrington presented a plea for some "hard-boiled realism" in the making of the peace and in the post-war world.

Officially titled "War on Wheels," the address which he presented at the Thursday evening banquet dealt only generally with this subject. His principal theme was the great responsibility resting on this Nation, not only in the winning of the war itself but in becoming the leaders and protectors of free peoples of the world after it is won.

"Our national pacifist tendencies of the past 20 years have truly brought us to the brink of disaster. Do we realize the gravity of our position? Are the American people fully aware of the implications of the events of today? What is the cause of this internal bickering, discussion, and seeming lack of unity of purpose?

"We are in this war to an extent which few of us realize. It is *our* war. If we fail to do our part it will be a lost cause. We must stop to realize that our British allies took a terrible beating in the last war.

"They lost over 1,000,000 men who would now have been between the ages of 40 and 50, and would have been the public leaders of today. This loss is going to impose a greater burden of the leadership in this war on us. Are we capable of assuming this leadership?

"We have got to be realists – hard-boiled realists – and we have got to take our place as the leading military power of the world if peace is to be maintained. We are vulnerable to airborne attack and this could be done on a small scale now – and probably soon will be. Consider five more years of advancement in the field of air transport and it is perfectly plain that our geographical position is no longer a factor in our national security."

President Herrington briefly and in general terms assured his audience that our production of war materials is beyond anything reasonably hoped for a year ago and that performance in the field of our tanks, trucks, and planes, was the equal of anything now in combat and superior to much opposed to it.

C. B. Veal, secretary of the Cooperative Research Council, in a paper "Cooperative Research Comes of Age," reviewed the steps of 21 years through which the joint research efforts of the SAE and the American Petroleum Institute have culminated in the creation of the comprehensive Council. He said the purpose of the organization was "to direct cooperative research in developing the best combination of fuels, lubricants, and equipment powered by internal-combustion engines."

"As to the limitations of this research," he stated, "standardization of methods of test, specifications, and classification shall not be within the province of the Council. It is the intent of the Council that such matters shall be promulgated by appropriate existing agencies."

With respect to organization, Mr. Veal said, "The American Petroleum Institute and the Society of Automotive Engineers govern the Council. Cooperating in supplying funds or services are the Automobile Manufacturers Association, the Aeronautical Chamber of Commerce, the National Bureau of Standards, and other technical societies such as the American Society for Testing Materials."

"Council membership is 12, appointed for one year, six by each of the two governing bodies. Council members must be empowered to speak authoritatively for their companies on engineering policies, must be active in performing their functions, and may not send proxies to Council meetings. SAE Past-President B. B. Bachman is the first chairman and Dr. T. G. Delbridge the first vice-chairman. The Council's functions are administrative, self-limited, primarily to business phases."

Work with armed forces

Mr. Veal illustrated the Council's functioning by telling of some of its work with the armed services, of the problems presented for solution, and how the various research groups of the organization had supplied data resulting in improved operation of service vehicles under extreme field conditions.

Discussing the meteorological conditions under which aircraft operate and so the conditions which rule the specifications of fuels and lubricants, Frank D. Klein, Standard Oil Co. of N. J., pointed out that aircraft operate under a wide range of atmospheric pressures, varying from sea level to less than one-quarter of sea level. Pressure variations influence chiefly the vapor-locking tendencies of the fuel system. Solubility of air in hydraulic oil and in aircraft engine lubricating oil also introduces problems with wide fluctuation of atmospheric pressure.

According to data on temperature of the upper air, which have been tabulated by AiResearch Manufacturing Co., the lower level of the stratosphere is at about 55,000 ft over the equator and at about 30,000 ft

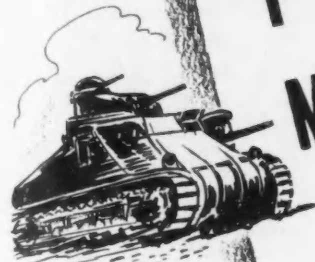
turn to page 24

These men made the Tulsa meeting "click" . . .

Among those especially responsible for the smooth operation of the 1942 SAE National Fuels and Lubricants Meeting at Tulsa were B. E. Sibley, Continental Oil Co., general chairman on arrangements; W. F. Lowe, Natural Gasoline Association of America, local chairman; with J. H. Baird, Lubri-Zol Sales Co., Carl Tangner, Diesel Power & Machinery Co., Tom Schuetz, Braden Winch Co., W. W. Scheumann, Cities Service Oil Co., and C. S. Hansen, Phillips Petroleum Co., as assistants.

THE SAE 1943 ANNUAL MEETING WILL BE

... SAE WAR ENGINEERING PRODUCTION MEETING



AND ENGINEERING DISPLAY

DETROIT

ARMY and INDUSTRY TECHNICIANS WILL SPEAK

TOPICS INCLUDE:

- Steel Cartridge Cases
- Servicing Army Vehicles
- Army Fuels & Lubricants
- Flash Welding in Aircraft
- Accessory Power for Aircraft
- Engineering Lessons, World War II
- Tractor Industry and the War
- Surface Finishes of Guns
- Bearing Alloy Corrosion
- Diesel Cold Starting
- Substitute Materials

JANUARY 11-15 Inclusive

T. B. Rendel, British Air Ministry, left, chatting with Capt. W. L. Hardy, U. S. Air Forces, Dayton, and H. R. Grigsby, superintendent of transportation, Oklahoma Gas & Electric Co.

Fuels and Lubricants Meeting

continued from page 22

over the poles. Temperature of the stratosphere varies inversely with the height of the lower level above the earth's surface, being about −120 F over the equator, as compared with about −50 F over the poles.

Of greatest need in engine lubricating oil development is improved stability with consequent reduction in engine deposits. Detergent type oils have proved effective in improving aircraft cleanliness and reducing piston-ring sticking. The use of anti-oxidants might be desirable in the future.

Concerning aviation gasoline, maximum efforts are being made toward the increased production of 100-octane fuel, and the further improvement of anti-knock value above 100-octane. Many new alkylation and catalytic cracking plants are in operation and many more are under construction to augment the supply of high-octane fuels.

Government restrictions on octane number of base stocks for the lower octane fuels necessitate the use of about ½ cc of lead per gal in 73-octane fuel and about 1¾ cc per gal in 80-octane fuel. This has led to some increased corrosion difficulties in the case of light engines not designed for operation on leaded fuels.

Most of the grease lubrication of aircraft can now be accomplished with only two grades of grease – one, a low-temperature grease and the other, a high-temperature grease. The low-temperature grease is used chiefly for intermittently operated units, such as control bearings and gear boxes which are never subject to high temperatures. The high temperature grease is used for continuously operated units, such as generator bearings, which are subject to low temperatures only for short periods of time and which normally are kept warm.

Corrosion preventives

Corrosion preventives for aircraft fall into two general classes: One, for protection of exterior surfaces of parts, and the other for interior protection of engines. Corrosion preventives, in general, consist of a rust-inhibiting base incorporated in a liquid or solid vehicle, depending upon the desired method of application and the type of protective film needed for the particular storage conditions involved.

In the discussion of Mr. Klein's paper, he was asked for more detail concerning oil dilution systems, and he said it was common practice for planes at northern stations to use such systems to facilitate cold starting. Only part of the oil is diluted in these systems, the diluent being gasoline. As the oil warms up, it loses 80% of the gasoline within 15 min and all of it within 30 min. Dilution by this system has shown no injurious effects on engines. Mr. Klein also stated that one of the reasons why the Russians were so successful in the air last winter was because they had good cold-weather lubricants and the Germans were greatly handicapped by lack of them.

In reviewing the improvements made in aircraft servicing equipment resulting from expanded wartime operations, Charles W. McAllister, Sinclair Refining Co., emphasized the efficiency of new designs in "Requirements of Aircraft Oil Servicing Equipment for the War Effort." Mr. McAllister stressed the need for extreme flexibility and versatility in all types of servicing equipment, indicating that while these are requirements of war they will be just as essential in the post-war era with the great growth of air transportation. Simplicity of design, maneuverability, and streamlined plumbing were deemed primary considerations. One of the most pressing problems in wartime servicing was said to be speed of oil drainage and refilling, where quick changes of planes in the field during operations are required. Trucks to meet these conditions must be rugged enough to move quickly over rough terrain and be equipped with mechanical devices of simple design and certain operation.

Improving used oils

Gilbert K. Brower, American Airlines, discussed the re-refining of aircraft crankcase oils, pointing out numerous methods for improving the used oils and the limiting considerations which must be observed to produce a first-class oil. Although his paper is withheld from publication or published comment, he presented verbally a great deal of first-hand practical information, useful to his hearers and showing the important advances made in this field by the military services.

In the operation of gasoline engines, J. J. Mikita, Harry Levin, and H. R. Kichline, The Texas Co., have found that aldehydes were the cause of the bad odor often noticed, and that aldehyde determinations of the condensate resulting from passing a portion of the exhaust gas through low-temperature traps correlated with exhaust odor intensity as determined by smelling exhaust gas samples. It was further found from actual measurement of air and fuel supplied to the engine during deceleration that the ratio of residual gases in the cylinder to air and fuel induced was very high, thus seriously affecting combustion and greatly increasing the formation of aldehydes which are intermediate products of combustion.

Additional tests showed that lean idling air-fuel ratios reduced deceleration odors and that fuel volatility was an important factor in so far as it affected liquid fuel formation on the walls of the manifold. The work indicated that a reduction in odor intensity may also be obtained by raising the intake mixture temperature, cleaning deposits from inside of the intake manifold, or by cutting off fuel or ignition during deceleration.

In discussing the paper on exhaust odors, T. B. Rendel, British Air Ministry, commented on the observed effects of low-temperature operation which resulted in combination of some moisture from combustion with the aldehydes greatly to increase the odor.

Errol J. Gay, Ethyl Corp., said the Ethyl companies' experiences had shown that the objectionable odor was always accompanied by a dense grayish-white smoke, in confirmation of Mr. Mikita's findings, and he recounted their work with one of the larger city bus operators when a definite procedure was found adequate to reduce the odor to acceptable limits, namely:

1. Apply enough heat to inlet manifold to obtain mixture temperature in 120 to 160 F range. Above 160 to 175 F, power loss was encountered.
2. Clean inlet manifold.
3. Raise jacket water temperatures.
4. Normalize carburetor
   a. Set float levels 1/16 in. low.
   b. Set idle lean as possible without stalling, usually in the 12.5 to 13:1 range.

Fine-structure copper-lead bearings are much less susceptible to corrosion than are the coarse-grained type made earlier, Leonard Raymond of the Tide Water Associated Oil Co. reported while discussing "Bearings and Bearing Corrosion." This is especially true at crankcase temperatures of about 250 F and above, he stated; at 320 F practically all oils are very corrosive, although they reach this stage at different temperature levels, indicating that different oils have critical temperatures at different values.

J. K. Anthony and J. E. Wilkey, Cleveland Graphite Bronze Co., in a discussion, explained that they carry out, in addition to microscopic study, a semi-micro analysis on tested bearings, determining if lead has been removed from the bearing surface

turn to page 30

T. A. Rogers, Shell Development Co., co-author of one of the Tulsa Meeting papers, discussing a point with Leonard Raymond, Tide Water Associated Oil Co., right. Mr. Raymond was chairman of one session, speaker at another

W.E.B. Detroit!

continued from page 21

defensive and offensive engineering operations. Conservation of critical materials through effective use, and the satisfactory operation and maintenance of military automotive equipment, constitute the defensive phase. The offensive calls for design engineering, for improved construction and performance of automotive equipment, the development of new, better, or alternate materials. Less suited to classification, but of equal importance, are the Board's continuing work in solving new operating and maintenance problems as they are created by the expanding use of motorized military equipment, its marvelously constructive service in preparing or compiling pertinent engineering information never before available, and its unending effort to solve the host of problems raised by the Nation's engagement in a war which demands the utmost in mobility.

■ Transitional Steps

A review which, because of wartime censorship, necessarily is sketchy, and which covers only a meagre cross-section of the Board's accomplishments, still is sufficiently enlightening to present an idea of the functioning of this technological arm of the Nation's military services. Presenting the Board's work neither chronologically nor in order of importance, this review does serve to trace the transitional steps from the time the Board was called upon to alleviate certain civilian inconveniences which marked the borning period of the wartime economy of scarcity until the Board began its present full-time schedule of engineering service for Government and military.

An 11-page report by the Board, made as early as December, 1941, outlined the probable effects of the compulsory use of low-octane gasolines upon the design, operation, and maintenance of motor vehicles, both military and civilian. The Board recommended reduction of gasoline octane numbers only to 72, seeking thereby to obviate necessity for widespread changes in available equipment, to avoid serious operating difficulties, and even to prevent a substantial upturn in gasoline consumption.

A Board report on anti-freeze solutions gave the military desired information, and probably prevented seriously extensive damage to Army equipment by warning of the destructive nature of some solutions previously considered suitable for use. In this same field of cold-weather operation of motor vehicles, the Board later was asked to recommend changes in engines and motor vehicles to assure satisfactory starting and operation in temperatures down to 40 degrees below zero.

Preparation of preliminary recommendations for cold-starting served curiously to link the Board to the tail of a veritable Polar bear which promises to increase in size and activity. Satisfactory cold-starting and sub-zero operation of motor vehicles, it developed, are of primary interest not only to the U. S. Army, which openly states it is preparing to carry on the business of mechanized warfare in arctic regions, but to so many others that meetings of the Committee on Cold Starting have assumed convention proportions. So far as widespread interest and complications are involved, the Board never has had a more comprehensive and, the Army insists, a more important project.

Although lacking facilities for extensive field testing under conditions which even simulate the continuous sub-zero temperatures and chill atmospheres of the arctic, the Board has supplied a gratified Army with a series of progress reports and, for the winter of 1942, expedients in the way of recommendations, priming and heating kits which will function on all types of vehicles, and a promise. The promise is that as quickly as a satisfactory natural refrigerator can be found, probably not too far off in view of approaching winter, the Board will undertake exhaustive experiments designed to provide the Army in 1943 with motorized equipment which will operate satisfactorily in the most severe arctic conditions and which will make the U. S. Army the superior of any cold-weather force in the world. The Army has replied with a go-ahead signal and announcement that it will establish a test camp in Northern Canada this winter. The Board now is lining up manufacturers of motor vehicles and automotive equipment for the most complete winter-operating tests ever made.

Progress reports of the Committee on Cold Starting began with the somewhat naive statement that "heat is the best answer to the problem," then proceeded to recommend a number of aids to easy starting. These included methods of and equipment for preheating engines and batteries, for keeping batteries warm, the use of primers, and the dilution of crankcase oil to release chilled bearings. Recommendations were made also that equal attention be paid to winter protection of transmissions, axles, differentials, and transfer cases.

The Board created subcommittees to make simultaneous studies of the many different phases of the problem. These groups now are at work on a variety of gadgets from radiator to tail-light. One of the first undertakings was the development of heating and insulation for storage batteries. In this connection, a sideline expedition into the field of behaviorism of storage batteries in cold weather unearthed painfully obvious proof that this is an unexplored and uncharted territory. Location of the battery under the hood beside the motor, a practice just coming into use with civilian motor vehicles when war halted their production, proved to be a move in the right direction. The battery, closer to the source of heat, could be warmed before the motor was started, and kept warm thereafter, especially if preheaters and heating coils were combined with adequate insulation.

■ New Problems

Expeditious starting of engines in cold weather presented a problem which took the research workers back to the days of hand-priming, led to the development of primers of adequate capacity suitable for application to existing equipment. Dilution of crankcase oil opened a whole Pandora's box of new problems, established good and sufficient reasons for positive crankcase ventilation, created a research job upon which the Cooperative Research Council and a number of subcommittees are working.

Board committees experimented with various types of heaters, estimated necessary capacities, devised ways and means of applying the gadgets to all types of equipment. The aid of service and personnel experts was invited with an eye to making these cold-starting aids so easy and so certain of operation, as well as so securely attached, as to withstand even the frostbitten nonchalance of Army truck drivers. Training the Army personnel in the use of the equipment has proved to be another important aspect of the whole project.

Since the Board's work quickly developed the fact that

parts of motorized equipment other than engines and storage batteries need cold-weather protection, future efforts are to be devoted to the additional task. The Committee on Cold Starting has been recruited from corporations manufacturing automobiles, trucks, tractors, engines, stoves, heaters, primers, farm equipment, batteries, and from oil companies and the Army. The cooperation of other groups has been promised, among them the Cooperative Research Council, which is interested especially in fuels and the effect of diluents upon lubricating oils. It is probable that the committee will grow in size and activity, especially because of the importance the Army is attaching to this complicated project.

The speed and scope of the work of the Committee on Cold Starting, as well as the whole-hearted support its efforts have received from industry, are typical of the spirit of service generated by the War Engineering Board. Lack of facilities for field and performance testing in the comparatively mild American climate was a handicap but no barrier. The Army purposely selected 40 deg below zero in the knowledge that cold-room facilities provided nothing more severe. The Committee on Cold Starting made a survey of cold-rooms in the automotive and related industries, obtained permission to use them, and inaugurated a series of simultaneous tests. Equipment manufacturers lacking cold-room facilities of their own were privileged to use those made available to the committee, and thus were enabled to subject their products – batteries, heaters, primers, and other cold-starting aids to tests not otherwise possible. These tests led to the formulation of preliminary reports and recommendations which the Army followed in its own final tests of motor transport equipment and combat vehicles in cold-rooms. Both the committee and the Army are preparing for more exhaustive research this winter at the arctic camp in Canada, where temperatures and atmospheres will be adequately sub-zero for sufficient periods to develop all the desired engineering data.

Two-way problem, also involving abnormal temperatures, referred to the Board by the U. S. Army Signal Corps is study of the effects of extremes of heat and cold upon electrical equipment, such as dynamotors and other power generators, field communications equipment, and Signal Corps apparatus which might have to be used almost anywhere in the world. The Board has filed a progress report on this global temperature problem, presenting preliminary recommendations and suggesting the use of possible substitute materials.

■ Materials Conservation

Parallel project is the study of the conservation of materials in electrical equipment, such as ignition coils, condensers, field coils, contact points, and regulators. Preliminary investigation has revealed the possibility of such savings as 15 to 20% of the copper in ignition coils, equivalent conservation in other items chiefly by simplification and standardization.

Recently the Board undertook to aid the Army in conserving the metal in grease and oil seals used in motorized equipment. By compiling the experiences of manufacturers and consumers, it was possible to prove that the waste in metal already has been cut to the irreducible minimum; that attempts at further savings would result only in other, and larger, losses.

The Board continuously has endeavored to temper or to detour many other shortages of critical materials as they have developed, seeking both to curtail consumption of certain metals for the specific purpose of prolonging their availability for essential uses, and also to encourage the use initially of alternate, ultimately of substitute, materials. Research leading to recommendations for wider use of steel and of copper alloys in the manufacture of radiators for civilian automotive equipment developed an effective pattern for conservation of copper. The results of the work of the Committee on Copper were reflected in recommendations for drastic savings in the amount of copper used in radiators of both civilian and military equipment. It won the outspoken approval of the War Production Board, which thereafter called upon the War Engineering Board to prepare its general copper limitation order, and asked the aid of the automotive engineers in preparing other limitation orders.

■ Tangible Savings

Board study pointed savings of 30 to 50% in chrome and nickel by welding chrome-nickel heads to steel stems for the severe service requirements of exhaust valves. A Board report indicated that as much as 25 tons of the critical metals could be saved weekly at larger plants by salvaging grinding muds, and recommended that worn valves be reconditioned whenever possible.

Investigation of the possibilities of conserving tin and lead revealed that several pounds per vehicle could be saved by revising formulas, using substitute materials. To conserve the supply of music wire, which is employed in the manufacture of springs, it was recommended that it be restricted to springs operated through wide ranges of stress and to springs of primary importance to the functioning of parts; that elsewhere substitutes be used.

Another report revealed that savings of 90% in nickel used in automotive replacement parts could be made. To prolong the nation's supply of cork, the Board proposed the use of treated paper fibers, of millboard, and of combinations of these materials with cork and with synthetic rubbers.

Even larger savings were found after a Board study of the possibility of curtailing the consumption of chromium, cork, copper, nickel, tin, zinc, magnesium, aluminum, rubber, and other critical materials used in the manufacture of Army half-track personnel carriers and scout cars. With the engineering practicality which is an attribute of all Board efforts, committees and subcommittees went over these vehicles, part by part, seeking means of sparing metal without spoiling service. Subsequent reports proposed a diet of substitutes which it was estimated would, on a per vehicle basis, whittle 24 lb of critical materials from the transmission, better than 15 lb from the chassis, one to three lb from the engine, and 19 to 22 lb from accessories.

These highly successful attempts to conserve metal recently have been expanded upon a scale which contemplates the savings of millions of pounds yearly. Ordnance equipment, including tanks of all sizes and types, is being subjected to pruning operations which will be definitely effective. Through the aegis of the Board, metals conservation is being made a fundamental feature of specifications for equipment going into future production. The idea is to incorporate conservation in the very design, and the objective is shifting from conservation by elimination or substitution during production to conservation by advance plan.

Briefly, the plan is to avoid specifying the use of high-quality metal when lower grades will serve. Since this objective calls for getting down to fundamentals and getting at sources, the Board is straightlining the project by bringing together design engineers, metallurgists, and representatives of the Army for concentrated, cooperative study. The Army is expected to say exactly what it wants, the metallurgists to determine what available materials will serve the purpose within the limits of design and stresses which the designing engineers know best.

Already under way are separate studies of the transmission, transfer cases and final drive of the suspension, of the turret, of the tracks and of the miscellaneous parts of Army tanks. Subcommittees concentrate on each unit, going over it bolt by bolt, and part by part, to ascertain where extreme stresses demand first-choice materials, where low stresses permit the use of second-choice or alternate metals.

Rough idea of what is being achieved is given in an estimate by Lt.-Col. J. H. Frye, of the Office of Chief of Ordnance, that the Board's recommendations to date will produce savings of more than 7,000,000 lb of nickel, of nearly 2,000,000 lb of chromium, and of nearly 1,000,000 lb of molybdenum in the 1943 production of medium tanks alone. This effort at metals conservation merely begins with blueprints and part-by-part examination; it is carried through laboratory and manufacturing trials right up to extensive proving ground and performance testing under Ordnance Department direction.

Further to the end of producing more and better equipment from lower alloy metals, Board committees are cooperating in a concerted effort of steel makers, fabricators, and consumers, including the military, to expedite the development and production of the most usable steels available. Objective is to make these alternate "National Emergency" steels so generally available at all times that shut-downs for lack of raw materials can be avoided, assembly lines can be kept in operation by substituting materials when necessary, and satisfactory performance and service is assured after delivery.

■ Far-Reaching Results

Far-reaching results, many of which transcend both the Board and World War II, are expected of metals conservation, and especially of the development of the new wartime steels. Construction and manufacture of military automotive and aircraft equipment initially will be affected, products of other industries later. Much remains to be done, both by the War Engineering Board and by other interested organizations, but it is patent that even in the making of raw materials for war the objective now is offense. It is a change from using whatever materials are available to the wholesale production, in adequate quantity, of materials specifically for war purposes.

Illustrative of the major importance of Board projects and of its methods of operating is the dramatic story of the Board's work for the solution of the critical national problem of tire shortage. This work, not yet completed, was initiated under the Board's auspices and now is being carried on by the Automobile and Rubber Industries Tire Committee of which James C. Zeder, W.E.B. chairman, is chairman also. By coordinating and concentrating industrial research and by aiding Government in developing voluntary public efforts at tire conservation, this work on tires literally has transformed muddled confusion into orderly progress toward effective results.

Virtually every step toward tire conservation and development of wartime interim tires has had its inception in the meetings of the Automobile and Rubber Industries Tire Committee. Secrecy necessarily has surrounded the efforts to find satisfactory substitute tires and tire materials, but sufficient progress now has been made and the future appears sufficiently bright to permit of the revelation that substitute materials now undergoing extensive testing promise to be so satisfactory for the manufacture and recapping of tires as to keep motor vehicles rolling in essential services on American highways and definitely to remove the threat of immobility.

■ Seeking Substitutes

This work began with a request from the National Inventors' Council that the Board develop all available information on substitutes for rubber tires. The request came at a critical time. War in the Pacific had shut off imports of natural rubber. Use of domestic stocks was restricted to limited essential services and to the military. Synthetic rubber production was insignificant, the material untried. Tire equipment was wearing, could not easily be replaced. Many solutions for the problem were being proposed, even pressed, but, lacking the element of practicality, they contributed more to confusion than to progress.

It was necessary to face the fact that there was no available substitute for rubber tires which could be used without adversely affecting the war effort. A nation accustomed to unlimited travel by highway, with some sections entirely dependent upon automotive transportation, was threatened by malignant immobility. It was unmistakably apparent that unless something drastic could be done, highway travel shortly must cease, and that curtailment of highway transportation eventually would disturb war production and deliveries, ultimately handicap the military.

The Board initiated its work by requesting chief executives of corporations manufacturing tires to assign their leading research and development experts as members of a Board tire committee which would compile and analyze information on every type of material which conceivably could be used to expedite production of an interim tire. Wholehearted cooperation was forthcoming and the committee convened at Detroit in May, 1942. It spent two days plumbing the depths of the problem and reviewing the possibilities of every known material and device which could replace rubber tires. It devoted a third day to making preliminary plans for attacking the problem of shortage.

Information and data were gathered from the experiments and experiences, the research and findings, of the whole industrial world. Whenever the trail of a possible rubber substitute was discovered, experts having knowledge of the material were called upon for full details. Representatives of Standard Oil of New Jersey, DuPont, Dow Chemical, Monsanto, Thiokol, and of other corporations and industries, came to tell of their own products and to offer helpful suggestions. Not only was every pertinent research development of industry reviewed, but every rubber substitute and synthetic, every proposed alternative for rubber-tired wheels, was given eager consideration. The committee avoided all prejudices, maintained a hopefully open mind, with the idea that any substitute, however

impractical it might appear, conceivably could be the key to the whole problem.

Several initial decisions were reached. Extensive research must be prosecuted to find substitute materials which would provide tires to keep civilian automotive equipment operating, at least in essential services. The material must be developed without interfering with the war effort, including Federal Government plans for making Buna S rubber, and without consuming critical materials. Experiments, research, and tests should be carried on by industry, but on a coordinated and cooperative basis which would eliminate duplication of effort, make every step of progress known to all interested, and overcome the handicap of a shortage of trained research personnel.

Available information revealing that tire life is prolonged by reducing highway travel speeds, decision further was reached that the public should be asked to drive at speeds below 40 miles an hour. It was decided that scrap rubber should be collected for reclaiming, and that reclaiming facilities should be worked to capacity. It became evident that motor vehicles should be removed from the highways as a safety measure when tires wore dangerously thin and that, to this end, periodic inspection of tires was required.

These decisions were embodied in the form of Board recommendations which, with other data, formed the first of a series of progress reports made to the National Inventors' Council. The series now has become a seven-volume compilation of the only complete data on tire conservation and rubber tire substitutes anywhere available. The data and recommendations have served as the foundation of the Government's tire conservation program subsequently developed by the Baruch Committee, and of the operations of the War Production Board's new rubber authority.

The first progress report stated that industry would undertake tire research work cooperatively, eliminate parallel developments, make equipment and personnel widely available for concentration on coordinated research projects, distribute all basic information as rapidly as it was developed, and generally build a broad foundation of understanding. This initial report indicated the Board's belief that Thiokol warranted further investigation as recapping material, secondary butyl for manufacturing interim tires. The report recommended that any efforts to promote the use of wood and mechanical-suspension wheels should cease, preliminary tests being unpromising. The report also offered the services of the Board in finding ways and means of curtailing the consumption of rubber in the manufacture of military vehicles.

Extensive Research

After a series of meetings of the committee and subcommittees, many held in industrial cities throughout the country for the convenience of research workers and for mutual review of progress, a second report was filed with the Council. It presented findings reached in an exhaustive study of mechanical substitutes for rubber tires, especially wood wheels. The report supported with proof the committee's earlier suggestions that wood wheels were unsatisfactory. It submitted voluminous evidence that while these substitutes might serve as emergency spares, their continued use would be ruinous both to vehicles and to highways, and would make the operation of motor vehicles dangerous to life and property. Further it was explained that the six or more lb of steel used in manufacturing and tooling a wood wheel from which not more than 200 miles of service could be expected, better could be used to make 12 lb of interim tire capable of giving 16,000 miles of service. This report had the salutary effect of discouraging the introduction in Congress of legislation designed to provide for an appropriation of millions of dollars to be used in subsidizing the development and manufacture of wood wheels.

Exhaustive Tests

Hundreds of other substitute materials were being tested, the second report said, adding that some appeared to have real possibilities. Among the substitutes undergoing consideration the report listed pine tar, cotton linters, wheat glutens, vulcanized oils, polymers, ester gum, corn cob gum, polyvinyls, wax, asphalt, carpet piling, guayule resins, stearine pitch, cottonseed oil, corn oils, tung, nylon, phenol-formaldehyde, fatice, soy bean elastromers, cellulose, and numerous others. The report further reviewed the continuing research within industry, and indicated that experiments were being made with detachable treads fabricated from cotton duck and webbing, leather, fabrics, and other materials, and with tread rings made of steel, of wood, and of molded materials.

The third report announced progress in a lengthy series of laboratory and road tests of thiokol used as a tire-recapping material. It was said the tests were demonstrating the possibility of recapping tires to give additional wear of at least 4,000 to 4,500 miles, the equivalent of one year's service. Recapping methods also under development were said to promise to conserve not only metals, but enough recapping material to double the number of tires serviced.

The fourth report revealed that tires made from secondary butyl, a petroleum derivative, and now called "flexon" tires, were proving to be 40 to 60 per cent as satisfactory as first-grade natural rubber tires. On passenger cars driven at speeds not exceeding 40 mph, the report added, flexon tires promised service of 13,000 to 15,000 miles, a mileage which could be extended considerably by recapping.

The Automobile and Rubber Industries Committee now is functioning as a complete entity carrying on research work and tests which promise the production of substitute and interim tires more satisfactory than previously it was believed could be developed. The objective now is not merely an interim tire, but a good interim tire. The interim tire already has been made, but plans now under way call for further laboratory and road tests to permit different formulas to be tried and experiments to be made with various combinations of materials so that the interim tire not only will serve as an emergency substitute, but deliver surprisingly good service.

Tire and automobile manufacturers, oil and chemical companies, now are cooperating in this work, exchanging data and progress reports to expedite the progress of the whole effort, operating with a singleness of purpose, and meeting all expenses. It is now, and confidently, expected that before present tire equipment is worn out satisfactory recapping material will be available in reasonably generous supply, and that the interim tires will be ready for use in 1943. These substitute tires are to be made completely from non-critical materials, are to be sold at reasonable prices, and are expected to give surprisingly satisfactory performance. The only anticipated handicap is that,

equipped with these interim tires, motor vehicles must be operated at speeds no higher than 40 mph, preferably less.

While the remarkable progress of the committee's work has been unknown to the public, it has been closely and appreciatively followed by Government officials and other key men whose letters of thanks and of commendation have been frequent and numerous. They include congratulatory messages from Donald Nelson, of the War Production Board; Leon Henderson, of the Office of Price Administration; James Conant, of the Baruch Committee; Secretary of Commerce Jesse Jones, and others.

A chart prepared by the committee graphically to present the potential conservation of tires by reducing highway speeds received the personally-penned approval of President Roosevelt. Thereafter it achieved the added distinction of publication, news mention, or editorial comment in practically every newspaper in the United States. The committee's work and recommendations further are reflected in the official reduction of maximum highway speeds throughout the country, widespread publication of driving rules to extend tire mileage, nationwide collection of scrap rubber, forthcoming regular inspection of tires, and other practical steps at tire conservation.

Suggestions made by the Board that study should reveal ways and means of saving rubber in the construction of military motor vehicles, and of using synthetic and reclaimed rubbers to conserve natural rubber in automotive replacement parts, were approved by Government officials and have been carried out. Subsequent studies have led to recommendations that plastics, felts, waterproofed paper, and synthetics and substitutes replace natural rubber for use in bumpers, steering wheels, brake linings, clutch facings, weatherstripping, floor mats, crash pads, and other parts, so that the use of natural rubber can be limited to those comparatively few critical purposes for which no other material will serve.

Savings in the amount of rubber used for storage battery cases and separators also is the subject of current study. Use of wood and of composition material, the development of new materials and the application of synthetic hard rubbers, plastics, and glass, are being considered. Also, a new material for storage battery cases is being sought diligently, products so far tested proving inadequate to meet the demands of extremes of temperature.

In the prosecution of a wartime engineering program, it is impractical and frequently impossible, to assess the importance of each project. What appears to be trivial one day may become astonishingly important the next, and what may seem like an inconsequential undertaking at the time may prove later to have been crucial.

Among the projects of the War Engineering Board mentioned in this review may be one or more which turned America's steps down the road to victory in World War II. The Board, however, is not concerned with such aspects of its work. It has disciplined itself to weigh each problem referred to it solely upon its engineering merits, and to get on with the job, whatever it may be, as if the outcome of the war were dependent upon its completion.

Highway to Victory

concluded from page 17

Appearance not only interferes with accessibility, but also, in some cases, interferes with the safe operation of the vehicle. The so-called streamlined cab is a case in point. There are cabs today that defy a full-sized driver to sit up straight and look out the windshield. We, operators, had been under the impression the windshield was a functional part, but some designers have almost convinced us it is simply an artistic relief for the gentle curve that joins the front of the cab with the roof.

In other cabs, the driver cannot get his knees under the steering wheel to operate the controls. In some cases, drivers have found it necessary to support their feet with stacks of bricks to get them into proper position to reach the controls. We have enough complications with mechanics and I believe bricklayers should be kept out of the automotive maintenance business. Need I add that at least one company is doing a pretty good business manufacturing seats which make it possible to get a driver into the cab.

So, the man who uses and cares for trucks and buses has suddenly become important. It took a war to make him so, but nevertheless he has arrived. Management now depends upon him as never before. Management still regards him as a little queer because it does not understand some of the things he talks about. The plain truth is, he is talking about the same maintenance problems he has always talked about, but management has been just too busy to listen. Frankly, it looks like the Office of Defense Transportation has a twofold problem. First, to improve the knowledge and ability of maintenance men and, second, to educate management on the difference between use and abuse – something it has never fully understood as regards automotive transportation.

It looks like the vehicle manufacturers will have some surprises in store for fleet operators that few of us suspect. Shortages have been responsible for their learning a great deal about substitutes. Some of these will no doubt in the end prove to be better for our purpose than the original material. Shortages have also been responsible for increased capacity for producing raw materials. One of these is aluminum. With the increased capacity to produce aluminum, the price might come down after the war to a point where this material can be used in many places where up to now it has not been economically possible.

We have already found out that tires are better than we thought. We simply did not know how to use them. There will be improvements in the tires themselves – and this, coupled with our increased knowledge of the use and maintenance of tires, will represent a cost reduction worth cheering about.

Fuels will be better as a result of increased capacity. This will permit engine designers to produce better and more economical engines. Experience with combat vehicles indicate that the gear transmission and friction clutch as we know it today is doomed.

All of these things and more are in store for us as truck operators. We are attempting to improve maintenance to take full advantage of them. We hope we will lose none of our good fortune because manufacturers choose to ignore our knowledge of what highway vehicles are bought to do and our experience with the problems involved.

Fuels and Lubricants Meeting

continued from page 24

selectively and how deep such penetration may have gone.

Messrs. Anthony and Wilkey recommend further study of a three-layer type of bearing, backed by steel, an intermediate layer of silver, lead-bronze, or other supporting material, covered by a thin layer of lead, indium-lead, copper-tin, or other suitable material.

Following the same lead as Mr. Raymond, Mr. Mikita showed that fine-grained copper-lead bearings lost only 0.5 g weight or less at 260 to 300 F crankcase temperature, whereas under the same conditions coarse-structure copper-lead lost as much as 15 grams. R. J. S. Pigott, Gulf Research & Development Co., expressed doubt that these conclusions on the effect of grain size on corrosion susceptibility could be taken as confirmed by the data presented, pointing out various complicating and conflicting conditions which must be considered. Mr. Pigott also pointed out that, while indium coatings are good as corrosion preventives, they wear away after a time, and therefore must be considered as palliatives rather than cures, useful especially during break-in.

Evaluation of piston skirt deposits has taken a decided step forward in the development of light densitometer methods for measuring these deposits, H. R. Luck, T. A. Rogers, and A. G. Cattaneo, Shell Development Co., believe. By using an Eastman reflection and transmission densitometer, two equivalent light beams are employed, one passing through a calibrated light-absorption wedge, the other being reflected from the surface to be evaluated. Variation of the wedge matches the coloration and light density, and numerical values may be assigned to the deposits. Still another method is to obtain a picture of the entire piston surface by rolling it on a circle while exposing a small strip of the surface at a time to a film, on which the entire picture is recorded either in black and white or on a photochrome film, by the principle of the focal-plane shutter. This device permits recording not only the amount and kind of deposit, but, on photochrome film, the color of the deposits also.

Good general agreement has been obtained in cooperative tests on rating engine oils by the Underwood test, even in the face of numerous variations in details of test procedure, C. W. Georgi, of Quaker State Oil Refining Corp., reported. As chairman of the SAE bench test subcommittee, Mr. Georgi reported the results of tests on six reference oils in 20 cooperating laboratories. These laboratories found that without iron catalyst most of the oils were rated rather close together in most respects, whereas with the catalyst wide differences were found among oils. This test, carried out for 10 hr at 325 F with 0.01% Fe_2O_3 (as iron naphthenate) catalyst correlates well with 36-hr Chevrolet engine block tests. The Underwood test conditions which correlate with the Chevrolet engine tests best are:

Crankcase Temperature	325 F
Time	10 hr
Volume of Sample	1500 cc
Catalyst, Fe_2O_3 as Naphthenate	0.01%
Copper baffle	2 x 10 in.
Cadmium bearing	1
Oil pressure	10 psi

In both Underwood and Lauson engine tests, the importance of temperature on evaluation of results was stressed. Lauson tests cannot be used alone and unfailingly to evaluate oils or predict service performance unless supported by large-scale engine tests, Mr. Georgi reported. He and his subcommittee were complimented on the thoroughness and dispatch with which the work has been carried on.

In discussing the practical service angle of this problem, Floyd Patras, Southwest Greyhound Bus Lines, stated he has found that the final test of serviceability is to place oils in 10 or 20 units, carefully observing and recording results over a period of at least three months, before reaching a decision on the service performance of any individual oil.

Southwest Group Students' Debate Results in Decision for Negative

Debating teams from Oklahoma University and Oklahoma A. and M. College, at the closing session of the Tulsa meeting, wrestled manfully with the subject "Resolved: That the two-cycle diesel engine is to be preferred when compared with the four-cycle diesel engine for automotive purposes," resulting in a victory for the A. and M. team, coached by Prof. R. G. Hilligoss and defending the negative side of the question. This winning team included Jack Douglas, W. C. Buck and Eugene Bright, with Dale Hugley, alternate.

The Oklahoma U. team was coached by Prof. Lowell E. Haas, and included William Junod, Don Crisjohn, and Paul Browne. Judges for the debate were T. B. Rendel, British Air Ministry; Frank Settles, attorney; Tom Schuetz, Braden Winch Co.; Matt Fairlie, Sinclair Refining Co., and Capt. Leo Towers, U. S. Engineers.

The students were awarded each a copy of the current "SAE Handbook" as prizes for their efforts.

Substitutes Will Be Standard Materials, Tractor Men Told

ALTHOUGH the ingenuity of the industry is now being taxed to the utmost in meeting the rapid changes which are occurring in available materials and resulting production operations as a result of the heavy influx of substitute materials for those now being concentrated in manufacture of war equipment, speakers addressing the SAE Milwaukee Section Tractor Meeting at the Milwaukee Athletic Club Friday, Nov. 6, all stressed that the tractor industry, among others, would benefit, predicting that many of today's substitutes would be the standards of tomorrow.

The one-day meeting, which attracted nearly 200 tractor engineers, allied technicians, and guests, was held with the cooperation of the SAE Tractor and Industrial Activity Committee.

Local arrangements were in charge of the Milwaukee Section officers and chairmen, headed by N. F. Adamson, Twin Disc Clutch Co., chairman of the Section. Charles T. O'Harrow, Allis-Chalmers Mfg. Co., was meetings chairman. Session chairmen were L. S. Pfost, Massey-Harris Co., chairman of the Tractor and Industrial Activity Committee; J. B. Fisher, Waukesha Motor Co., who handled the afternoon meetings; and Mr. Adamson, who presided at the banquet and evening meeting.

Speakers were Hyman Bornstein, Deere & Co., and Dr. Carl E. Swartz, Cleveland Graphite Bronze Co., both addressing the afternoon sessions; A. C. Fiedler, Northwestern Mutual Life Insurance Co., and A. T. Colwell, Thompson Products, addressing the banquet and the evening session, respectively.

That many of the changes in alloy steels during the past two years would "become permanent," and that "we will have better alloy steels at lower costs when the war is over," was predicted by Mr. Bornstein in his discussion of "The Changing Picture in Alloy Steels," covering the numerous changes in the alloy steel picture during the past two years as a result of the war and its requirements for war production. Many of the substitutes, such as molybdenum, became scarcer than the originals, as for instance, nickel and chromium, because of the heavy drain on the alternate supplies after the nickel and chromium became unavailable to the industry. The result was the compilation of a new series of NE steels in August, which eliminated some of the original NE steels and the 4000 series as well.

"Some new steels have been added, and I want to call your attention to the 9400 series. You will note that the nickel and chromium contents are in the neighborhood of 0.30%," Mr. Bornstein commented. "In many instances, alloy mills can obtain this amount of nickel through residuals and the amount of chromium as residual will be in the neighborhood of 0.10%. Consequently, it would be necessary to add only a small amount of chromium and a small amount of molybdenum to produce this series of steels. We have not had much experience even in the laboratory way, with the 9400 series, but it does look as though it is a well-balanced series of steels."

Mr. Bornstein pointed out that the current NE list includes only the 1300, 9200, 9400, and 9600 series as available for the tractor industry, because the other steels contain more nickel, chromium, or molybdenum, "and it is desired to reserve these steels for strictly war uses such as tanks, airplanes, and so forth," he explained.

Commenting on the changing picture of hardening elements, Mr. Bornstein pointed out that the list six months ago would have shown carbon as the least critical, with silicon next, then manganese, molybdenum, chromium, nickel, and vanadium. The shift from nickel and chromium to molybdenum had made a considerable change in this arrangement and now molybdenum is much more critical than chromium or nickel. Mr. Bornstein stated that the WPB has taken steps to obtain a balanced situation.

Mr. Bornstein referred to the importance of hardenability in steels and the effect of chemistry on hardenability. He referred to the recent paper by M. Grossman of Carnegie-Illinois Steel Corp., which indicated that if the complete chemistry is known, the hardenability can be calculated within close limits. The paper further showed the considerable effect on hardenability of small amounts of various alloying elements.

Another development in the steel industry, which has become very important recently, is the use of special alloy-addition agents such as Grainal. The use of these special alloy-addition agents results in a considerable increase in hardenability and also in improving the toughness of the steel at high hardness. Mr. Bornstein predicted a greatly increased use of these addition agents in order to save on critical alloying elements.

Discussion following Mr. Bornstein's paper covered questions on machinability of some of the available steels, with the opinion being expressed that some adjustments are necessary in machine practices when using the new series of steels. No definite reactions have as yet been possible on most of the new series, primarily because of the comparatively short time which has elapsed for any extensive, thorough experiments. No particular complaints have been evident, although in some instances primary tests have shown that the available series have not been "on all fours" with the steels formerly used. The 8620 series steel has been found especially suitable for carbonized gears used in marine work. While the 1300 series has worked out well, discussion stressed that this series is not as fool proof as some of the original series, with the result that more care and supervision are required in its use. However, when properly used, it has proved its merit in both the tractor and automotive industries, and at a cheaper cost than some of the originals. One objection to the 1300 series has been its somewhat poorer machinability.

Changes in the bearing industry because of the war, while evident, have not been as drastic as in some other industries in that it has been necessary for bearing manufacturers to change only the character of the product and not the product itself, Dr. Swartz pointed out in his paper on "Material Substitutions and Developments in Engine Bearings."

"If we classify bearings in the groups: aircraft, automotive, diesel, and miscellaneous, we find that bearing production has increased in all fields except automotive," Dr. Swartz pointed out. "In the automotive group, we find that there has been an increased production in the larger sized bearings, indicating that manufacturers have increased their production in all fields except that which represented passenger-car bearings, formerly the largest field, but now decreased almost to the vanishing point."

Left to right, N. F. Adamson, Milwaukee Section chairman, SAE past-president A. T. Colwell, who spoke at the evening meeting, and C. T. O'Harrow, Milwaukee meetings committee chairman, who was responsible for arrangements

Substitution of lead-base bearing alloys for tin-base babbitt, influence of the ultra-thin linings on bearing life, and the evaluation of some of the newer bearing materials, were discussed by Dr. Swartz in his paper. He stressed that the tin-base babbitts are now used only on express order of the armed forces.

A suitable substitute for the tin-base bearings is F-23, an alloy which has been under test for three years, and which has given a performance considerably more uniform than SAE 14, developed in 1936, and equal to that of the tin-base babbitts, Dr. Swartz explained. He stated further, "I can recommend F-23 not only as a low-tin alloy for use during the war period, but as an alloy for consideration when tin again becomes more plentiful. Briefly, its properties are equal to those of the best tin-base babbitt; its hardness is retained over a long period of time when exposed to elevated temperatures. Softening at elevated temperatures is one of the weaknesses of alloys of the lead-tin-antimony system that is corrected when arsenic is added; fatigue strength is superior to tin-base babbitt and other lead-base alloys; and corrosion resistance is equal to that of tin-base babbitt and superior to some lead-base alloys."

Discussing the trimetal type of construction for bearings, Dr. Swartz said that "from a metallurgist's viewpoint, this type of construction seems a logical one. Three metallic layers are put together. Each has its particular contribution to make, and the result is a product whose properties are difficult to get from any single material."

The new trimetal bearing, as well as the silver and aluminum bearing, will be heard from more in days to come, Dr. Swartz predicted, stating that silver bearings have been used in the aircraft industry for some time, and since the silver has been coated with lead and then plated with a thin layer of indium to prevent corrosion, this bearing has given excellent results, with "loadings over 10,000 psi of projected area being reported in satisfactory prolonged operation." Commercial use after the war, however, would be limited because of the high cost of silver.

While England had the aluminum bearing in use and under test up to the outbreak of the war, this bearing has not been widely adopted. Tests made in this country show promise for this type of bearing.

"One of the problems that the designer will have to cope with in the use of aluminum bearings is the high coefficient of expansion of the aluminum alloy which will prevent, for example, the use of the present conventional method of anchoring bearings.

"Both the aluminum and the silver have the advantage of a low modulus of elasticity of the order of 10,000,000 psi and both have the advantage of reasonably high fatigue values. At present the silver bearing needs a lead coating, and it may be that aluminum will also. Neither has the surface or bearing characteristics that tin and lead alloys have."

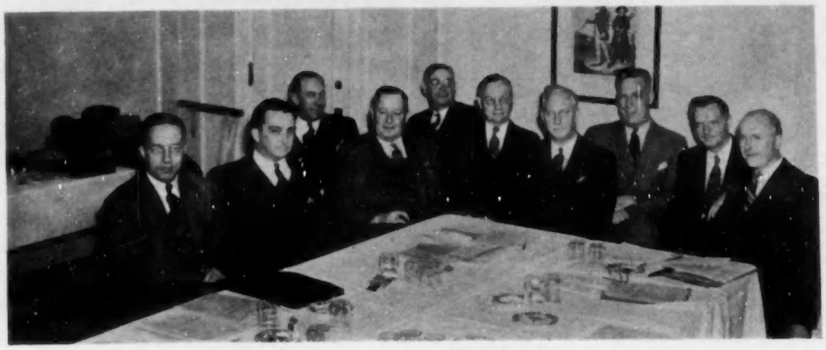
Curtailed production of tractors, corn planters, reapers, and other farm equipment, plus the shortage of repair parts, have placed "used" farm equipment at premium prices, according to A. C. Fiedler who spoke on "The Farm Equipment Situation from a Farm Operator's Point of View." Basing his facts on his recent experiences in the field, he pointed to the high prices which used equipment is bringing at farm auctions. For instance, a 1942 tractor, originally priced at \$1100 sold for \$1500; a 3-year old combine costing \$600 new, sold at \$1100; a \$550 corn picker brought \$850.

This shortage of new and used equipment, plus the steady drain of efficient, trained farm help, which has taken 1,300,000 workers from farms to war industries so far this year, compared with 570,000 in 1940, will

Future – Present

C. G. Krieger, Tractor vice-presidential nominee for 1943, talks with L. S. Pfost, right, now SAE vice-president representing Tractor and Industrial activity

SAE Tractor War Emergency Committee



The SAE Tractor War Emergency Committee, of which A. W. Lavers, Minneapolis Moline Power Implement Company, is chairman, at its meeting in Milwaukee on Nov. 6. This committee is currently rendering engineering advisory service for the WPB and the OPA and is organized to work on war problems having to do with tractor and farm machinery engineering

make it extremely difficult for farmers to cooperate in the Government's request for greater production, Mr. Fiedler stressed.

American-designed and manufactured planes were placed on a par with "anything" any other nation in the world has to offer, by Mr. Colwell who discussed "Valves For Wartime Requirements." He said criticisms against our plane designers and manufacturers by some persons "writing for money" were unjust and untrue, and were definitely a morale reducer among the general public which does not know all of the facts. He pointed to the great success of our planes used by the Flying Tigers, and a more recent example, the 7 to 1 ratio in our favor in September air wars in Europe.

In his talk, Mr. Colwell also covered several of the new developments in our aircraft for the armed forces, predicting that more innovations will further enhance the superiority of our planes over those now operated by the enemy. He stressed that our Government has deemed it better to slow down the speed and reduce the maneuverability of planes by including armor and larger guns, than to build ships which offer a minimum of protection to the fliers, such as the Jap Zero planes do.

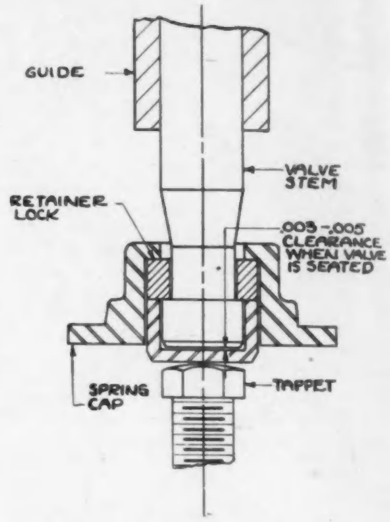
Discussing the trends in valve construction, Mr. Colwell predicted that some of the substitutes now being used would continue in use after the war, with many of the "originals" never to come back into the production picture again.

Mr. Colwell then gave a description of the recently developed "rotating" valve, which has been designed to prolong valve life in automotive and aircraft engines operating under extreme service conditions.

A special locking device at the tip, as shown in the accompanying illustration, permits the valve to rotate slowly in the guide without interfering with normal operation, producing a light lapping action. Seat and stem deposits are thus removed before the accumulation at these points can cause blowby or sticking.

The principle of the valve is to free the valve stem from the retainer as it is opened, so that forces inherent in the valve train can induce slow rotation.

Mr. Colwell mentioned that exhaustive dynamometer and road tests have proved that this design will increase valve life several times. He told of one example of valves which formerly gave only 100-hr service under adverse conditions in a power unit and are giving more than 1000 hr with this design; and another instance where service was increased in a popular truck engine from 5000 to 60,000 miles between valve replacements.



Detail of rotating valve retainer assembly described by A. T. Colwell in his talk at the Milwaukee Section Tractor Meeting on Nov. 6

SAE Coming Events

Jan. 11-15 1943 — **SAE War Engineering Production Meeting (and Engineering Display) Book-Cadillac Hotel—Detroit, Mich.**

Baltimore – Dec. 3

Engineers Club; dinner 6:30 p.m. Maintenance from a War Time Angle – W. J. Cumming, Chief of Vehicle Maintenance Section, Division of Motor Transport, Office of Defense Transportation.

Buffalo – Dec. 9

Markeen Hotel; dinner 6:30 p.m. Modern Pipe Lines for Natural Gas – Henry Lehn, consulting engineer, Worthington Pump & Machinery Corp. Some Problems of Pump Applications to Oil Pipe Lines – V. Gerbereaux, assistant manager, Centrifugal Pump Division, Worthington Pump & Machinery Corp.

Chicago – Dec. 8 and 9

Knickerbocker Hotel. Air Cargo Engineering Meeting.

Cleveland – Dec. 14

Cleveland Club; dinner 6:30 p.m. War on Wheels – A. W. Herrington, chairman of the board, Marmon-Herrington Co., Inc., and president, SAE.

Colorado Club – Dec. 8

Blue Parrot Inn, Denver; dinner 6:00 p.m. Power Farming – M. S. Anderson, tractor and power plant salesman, McCarty-Sherman Motor Co.

Detroit – Dec. 7

Horace H. Rackham Educational Memorial Building; dinner 6:30 p.m. Machineability of War Materials – Prof. O. W. Boston, University of Michigan. Gun Design as Related to Use – Col. H. W. Miller, University of Michigan.

Indiana – Dec. 10

Antlers Hotel, Indianapolis; dinner 6:45 p.m. Ground-Crewing "The Flying Tigers" in China – Tye M. Lett, Jr., Allison Division, General Motors Corp.

Metropolitan – Dec. 10

Hotel Edison, New York City; dinner 6:30 p.m. Tank Power Plants – Major Howard R. Hammond, U. S. Armored Force, Fort Knox.

***Milwaukee* – Dec. 4**

Milwaukee Athletic Club; dinner 6:30 p.m. Speaker – J. J. Frey, Ethyl Corp.

***New England* – Dec. 10**

Engineers Club, Boston; dinner 6:30 p.m. Subject – Synthetic Rubber. Speaker to be announced.

***Northern California* – Dec. 8**

Palace Hotel, San Francisco; dinner 7:00 p.m. Turbocharging 4-Cycle Diesel Engines – J. P. Stewart, manager, Supercharging Dept., Elliott Co.

***Oregon* – Dec. 11**

Lloyds Golf Course, Portland; dinner 6:30 p.m. Symposium – Reclamation of Automotive Parts.

***Peoria Group* – Dec. 28**

Dinner Meeting for SAE members only. Round table discussion – "Design Problems of Diesel Engines."

***Southern California* – Dec. 11**

Hollywood Roosevelt Hotel, Los Angeles; dinner 6:30 p.m. Junior Activity Meeting. Speaker – H. E. North, research engineer, Douglas Aircraft Co., Inc.

***Southwest Group* – Dec. 4**

Tulsa. Metallizing – Charles K. Stipp, district manager, Metallizing Engineering Co., Inc.

University of Texas Expands Aero Courses

The University of Texas recently announced the appointment of Dr. M. J. Thompson as chairman of the newly created Department of Aeronautical Engineering. This work in the aeronautical field is an outgrowth of courses started in 1926 and previously offered in the Department of Mechanical Engineering. An extensive program of engineering, science, and management war training courses is also being carried on in the aircraft field, with courses available in Austin, Fort Worth, Dallas, Houston, and other points throughout the state.

Dr. Thompson studied extensively in aeronautical engineering in the United States and Europe, and holds degrees from the University of Michigan and the Warsaw Polytechnical Institute. He has also served as a consultant to numerous aircraft and other industrial organizations on problems in applied aerodynamics.

Staff Members

Assisting Dr. Thompson in this work are Dr. M. V. Barton, formerly a member of the teaching staff of Cornell University and the University of Maryland; Harry W. Brown, a graduate of the University of Nebraska, former holder of a Boeing scholarship in aeronautical engineering, and member of the engineering staff of the Vega Aircraft Corp.; and John C. Bowman, Stanford University graduate, formerly with the Consolidated Aircraft Corp. in San Diego, who is in charge of the engineering, science, and management war training aeronautical program in Fort Worth and Dallas.

The University plans to expand both its instructional and research work in the aeronautical field to meet the tremendous demands for trained technical personnel.

Chicago Section

AIR CARGO ENGINEERING MEETING

December 8 & 9, Hotel Knickerbocker, Chicago

■ PROGRAM ■

TUESDAY, DECEMBER 8

10 00 A.M.

William C. Littlewood, Chairman

Deficiencies of Converted Passenger Airplanes for Cargo Operation Requirements
– Charles Froesch, Eastern Air Lines

Terminal Handling of Air Cargo
– Karl Larson, Northwest Airlines

2:00 P.M.

Hon. William A. Burden, Chairman

Packaging and Handling of Air Cargo
– C. G. Peterson, Railway Express Agency

Some Aspects of Air Cargo Operations in Latin America
– J. Parker Van Zandt, Office of Air Transport Information, Department of Commerce

The Economics of Post-War Carriage of Air Cargo
– J. V. Sheehan, Lockheed Aircraft Corp.

WEDNESDAY, DECEMBER 9

9:00 A.M.

Peter Altman, Chairman

Securing Means for Air Cargo
– E. S. Evans, Evans Products Co.

Structural Materials for the Cargo Airplane
– H. D. Hoekstra, Civil Aeronautics Administration

Airplane Design for Cargo Transportation
– Carlos Wood, Douglas Aircraft Corp.

2:00 P.M.

C. Graddick, Chairman

Air Pickup and Gliders As Related to the Future of Air Cargo
– Richard DuPont, All American Aviation, Inc.

Gliders for Transport
– Major L. D. Barringer, Chief, Glider Unit, Directorate of Air Support, Army Air Forces.

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Dinner

Tuesday, December 8 **6:30 P.M.**

R. D. Kelly, United Air Lines
Chairman

W. B. Stout, Stout Skycraft Corp.
Toastmaster

SAE War Effort Activities
A. W. Herrington,
President, SAE

Naval Air Transport Service
Com. C. H. Schildhauer,
Naval Air Transportation Service

The Importance of Air Cargo in the War Effort
Col. Harold R. Harris,
Assistant Chief of Staff, Air Transport Command,
Army Air Forces

This meeting is under the auspices of the Chicago Section of the SAE, with the cooperation of the SAE Aircraft Activity, the Air Transport Association of America, the Aeronautical Chamber of Commerce of America, and the U. S. Department of Commerce.

Illustrated Instructions Make Repair Jobs Easier

▪ *Southern California*

"Throughout the aircraft industry there is a great need for information on the operation of aircraft functional systems and parts," declared George Tharratt, chief engineer of the Adel Precision Products Corp., in his talk on "Education of Air Force Service Men" at the Oct. 16 meeting of the Southern California Section. "Because of the widespread shortage of skilled labor now," he continued, "this need has become urgent in all factories."

Mr. Tharratt explained that the only available material of an educational nature is found to a limited extent in handbooks, prepared to Army Air Forces specifications as technical orders. Thousands of aircraft employees, to whom these technical orders are unavailable, require material on functional operations for the efficient execution of their jobs.

The "Production Illustration" method, which has been used by several companies and has proved its worth, was described by Mr. Tharratt. "Written material in contrast to illustrative material," he said, "requires mental interpretation. Although engineering blueprints are not written material, they do require visualization. These intermediate steps are often the cause of much misinterpretation and loss of time," Mr. Tharratt said. "The new function illustrations will present this material in pictorial form, clearly and directly," he declared.

In closing Mr. Tharratt stated that all this knowledge is free for use by the United Nations, and no patents are involved.

The next speaker was Stanley A. Wilson, division manager of Pacific Aeromotive, Lockheed Air Terminal, who delivered a paper on "Aircraft Maintenance."

"Past standards in maintenance and service industries were largely based on training, at a great length of time, highly competent and thoroughly schooled mechanics who literally grew up with the aircraft industry," Mr. Wilson said. The process was a long and tedious one, and often the mechanic made his own parts and was generally capable of performing all operations on the aircraft, such as fabric covering, repair, patching, welding, and so on, he pointed out.

With the advent of war conditions, and particularly now that production is well under way and operations have reached an unpredictable degree of flying hours, he continued, a vast change in aircraft, engine, propeller, instruments, and accessory maintenance service and overhaul has been in process. Aircraft maintenance and overhaul have evolved into highly specialized operations wherein an individual can be trained to fit piston rings to a specific model of engine within a few weeks' training, for example, he pointed out. Now each individual is a specialist at his or her endeavor. We here in the United States may have some real service problems, but we are doing all that we possibly can in production of new aircraft and repairing aircraft without delay. When we consider the conditions under which repair jobs are being done at the frontier combat zones, we realize that our conditions here are somewhat more ideal, Mr. Wilson emphasized.

Army Service Methods

Major A. W. Ries, 205th Artillery at Fort Lewis, Washington, now serving with the Artillery in Southern California, gave a talk on "Army Service Methods." Major Ries told how operating in the field without shop equipment necessitates unit replacements. Sometimes it may take six weeks before the parts are obtained, which means that we have many vehicles tied up at times because of lack of parts or units, he explained. "However," he added, "more good tools and equipment are coming through now." It has been found, he said, that only by repetition is it possible to train drivers properly to inspect their vehicles and understand the great importance of these frequent inspections. Their records show that 187,000 miles were covered in one month in Los Angeles without even scratching a fender – and that required training, he declared.

Master Sgt. Allen G. Fisher continued the discussion on army maintenance. He explained that repairs are being made under the worst conditions of dirt and sand, and being successfully done by novices. "If you design any repair equipment," he pointed out, "let it be simple and strong."

Phiroze Nazir, aeronautical research engineer, H. M. Government of India, who is in this country to present his revolutionary cut-slot wing for airplanes, said, "I am happy to give my services in any way to the United Nations," and remarked that we must forget our individual problems and bring concerted energies to bear against the Axis.

Cites Similarities of American and Jap Engines

▪ *Pittsburgh*

The Pittsburgh Section of the SAE and the Aero Club of Pittsburgh sponsored a joint meeting on Oct. 27, at which W. G. Ovens, staff engineer of the Wright Aeronautical Corp., presented "Some Notes on Design Features of the Mitsubishi Kinsei Engine." This paper was published in the July issue of the *SAE Journal*, pp. 253-266.

Gus Haller of the Aero Club started the discussion by pointing out some of the resemblances in design between the Japanese engine and engines of American make. Mr. Ovens replied that "who copied whom" would always be largely a matter of opinion, permitting almost endless discussion.

"Cannon on Wings," colored sound pictures of the Airacobra Pursuit Ship by the Bell Aircraft Corp., were shown.

The Pittsburgh Section members were guests of the Aero Club at a cocktail period preceding the dinner.

Causes and Corrections for Copper-Lead Bearing Failure

▪ *Philadelphia*

"Copper-lead bearings are subject to three types of failure – selective loss of lead through high temperature corrosion, loss of both copper and lead through low temperature corrosion, and structural fatigue," declared Dr. J. C. Geniesse, research engineer of the Atlantic Refining Co., when he presented his paper, "Bearing Failures in Commercial Vehicles," at the Oct. 14 meeting of the Philadelphia Section.

Making use of colored slides of bearing sections, Dr. Geniesse analyzed in detail the various types of failure which have been experienced with this most popular of new type alloy bearings. Ninety-five per cent of the slides were made from bearings out of actual service, he said.

"The type of structure," Dr. Geniesse pointed out, "has a decided effect on the susceptibility of the bearing to failure, both from corrosion and from fatigue. The fine-structured, evenly-distributed mixture is much better than the coarser type."

Slides shown by Dr. Geniesse to illustrate the types of failure were prepared by sectioning the bearing at the point where failure appeared to occur. This section was then molded in a bakelite block and the surface polished and etched in preparation for the microphotograph. In this way the complete thickness of the bearing, and not merely the surface, is open to inspection.

Shows Slides from Engines

The first type of failure, that caused by high temperature corrosion, was illustrated by a number of slides from both gasoline and diesel engines. "Acids formed through high temperature deterioration of the lubricating oil have selectively attacked the lead along the surface of the bearing," Dr. Geniesse explained. "The porous layer of copper which remains either compresses or disintegrates, the clearance of the bearing increases, and the engine becomes noisy," he continued. The cure for this condition is the elimination of the deterioration by "maintaining lower oil temperature" or by properly "inhibiting the oil against deterioration" and by "using proper drain periods."

The second type of failure – low temperature corrosion – was illustrated by slides


The Oregon Section of the SAE held its Oct. 9 meeting at the Consolidated Freightways Shops in Portland. The complete shop installations and maintenance procedure was given a thorough inspection. All of the engine chassis maintenance is handled in these shops, where they have complete equipment for doing all types of repair work.

showing attack, not only of the lead, but of the copper as well. Thus the complete surface is eaten away, sometimes as deep as one-half the thickness of the bearing material. "The cause of this type of corrosion," said Dr. Geniesse, "is attack by corrosive acids introduced in the fuel and collected in the crankcase because of poor ventilation and low crankcase operating temperatures." The most satisfactory remedy is, of course, "to increase the crankcase ventilation," he said.

The third type of failure, structural fatigue, is characterized by cracking of the bearing surface. The cracks were shown in a good many cases to extend well down toward the steel back of the bearing. The causes of this type of failure are sometimes very difficult to determine. "Some of the common causes," said Dr. Geniesse, "are excessive speeds, excessive loads, and in a great many cases improper installation. Copper-lead bearings must have more clearance than the older type babbitt bearings."

Dr. Geniesse, as an oil man, expressed a willingness to accept a part of the responsibility for the bearing failure problem. He believes, however, that the engine manufacturer, the vehicle operator, and the service man must also accept their share.

Mitsubishi Kinsei Engine Is Analyzed

▪ *Milwaukee*

W. G. Ovens, staff engineer of the Wright Aeronautical Corp., was guest speaker at the Milwaukee Section's Sept. 26 meeting. Four hundred people came to hear Mr. Ovens present his paper, "Some Notes on Design Features of the Mitsubishi Kinsei Engine." This paper appeared in full in the July issue of the *SAE Journal*, pp. 253-266.

Tour Conducted at Holabird Motor Base

▪ *Baltimore*

The Baltimore Section's Oct. 15 meeting consisted of a tour of the Ordnance Motor Base, Holabird, Baltimore. The members and guests were conducted through the paint laboratory, where they were shown some of the experimental equipment in the engineering department; the compact tire retreading and capping plant; and the main repair shop's rebuilding of motors and grinding of crankshafts.

The next step in the tour was a visit to the school where Army students receive the fundamental training on chassis, carburetor, electrical clutch transmission and rear assemblies, and the light machine shop where the students are taught the use of small lathes, shapers, presses, and portable crankshaft grinders. From there they were taken to the body shop where repairs to body fenders, tanks, and radiators were being made by Army students.

A meal prepared by the Army cooks was served in the mess hall, and then they were conducted to the new auditorium. The meeting was opened by Col. Herbert T. Lawes, commandant of school. Major Kelley, assistant commandant of school, Col. Airs, chief of the supply division, and Col. Bieglow, chief of technical service, spoke briefly. Col. Johnson, chief of engineering, told of the problems confronting his department, and the work being done in connection with civilian engineering. Movies of the Army's accomplishments were shown.

Herrington Speaks at Indiana Meeting

▪ *Indiana*

"Let's emulate our Russian allies and do less talking and get on with the job of killing more of the enemy," warned SAE President A. W. Herrington at a meeting of the Indiana Section on Oct. 15 at the Antlers Hotel in Indianapolis.

He described this war as ". . . a war of the people in which peaceful nations fight for their very existence, against the predatory desires of aggressor nations. These aggressors have demonstrated their desire not only to accomplish economic gains, but to enslave the conquered peoples, and to justify their action by assertions of racial superiority.

"Our national pacifist tendencies of the past 20 years have truly brought us to the brink of national disaster," he continued. "We have got to join with our worthy allies, China, England, and Russia, and do far more than just act as the arsenal of supplies," he said.

John A. C. Warner, secretary and general manager of the SAE, told of the "SAE's Part in and for the Armed Forces," and two motion pictures were shown – Lockheed's "Leadership," and Martin's "Building a Bomber."

Lt.-Col. Cummings Ill

The meeting of the Metropolitan Section scheduled for Nov. 19 was canceled due to the illness of Lt.-Col. Carl E. Cummings, who was to speak on Current Tank Design.

Old-Fashioned Parts Packing Inadequate for War Needs

▪ *Canadian*

"Years of experience in the automotive vehicle business lead those in the industry who are charged with the responsibility for supplying material to believe that it is not possible to deliver the thousands of tons of spare parts that have been sent to the fighting forces, without receiving at least a few complaints. So, largely upon their own initiative, the manufacturers have gone to great pains and expense to learn the exact condition of material when received at destination," R. A. Shelly, parts war contracts manager of the Chrysler Corp. of Canada, Ltd., revealed at the Canadian Section's meeting on Oct. 28 at the Royal York Hotel in Toronto. He then proceeded to tell what the investigations have shown, and what the industry is doing to remedy these evils.

"We are now receiving some information from the Army overseas," he continued, "and the accumulation of all this information has shown us that the situation to date has seriously affected vehicle operation, and has been extremely wasteful. To continue to pack material in the old-style way, knowing that only a relatively small percentage of it would be useful upon arrival at destination, would be sabotage."

Problems in the automotive industry come up fast, but they are knocked down fast, Mr. Shelly declared. He illustrated new packing methods by the use of slides. While this packing procedure cannot claim to be perfect, it is far superior to anything that has been used heretofore, he pointed out.

For example, several steps must be followed in the correct packing of valves in order to insure them against rust and corrosion. He pictured these steps with a film which showed, first, the valve being dipped in a special rust-proofing material which drys fairly soft – it is about the consistency of a heavy grease – and which will unite with lubricating oil. Thus the washing process is eliminated, as it is only necessary to wipe off the superfluous rust-proofing material and install the valve.

His next slide showed the valve being wrapped in a special waterproof waxed cloth, in such a way that there are no folds which permit easy entrance of water or water vapor.

The third step, he continued, is the sealing of the valve which has been wrapped in

the waterproof cloth. The wrapped valve is dipped into a bath of hot sealing compound which dries to the consistency of a hard wax. Then the sealed valves are placed upon a wet pad which causes the wax to congeal – an idea borrowed from the candy maker who places his chocolates on a wet pad to prevent the chocolate from dripping.

Finally, each valve is placed in its own individual carton, thoroughly protected against rust and corrosion, and a sheet of waterproof cloth is placed on top of the parts, and the flaps of the bag liner sealed. These films are an example of the number of small cartons that are required for each pack of parts.

Mr. Shelly explained that experimental work and testing are being carried on continuously and nearly every day minor changes are made. He hopes that ultimately the combined knowledge and resources of the industry will devise a method of packing which will be practically perfect.

Collaborating with Mr. Shelly was Jack E. Harper, boxing engineer of Ford of Canada, who presented a technicolor motion picture which revealed the irreparably corroded condition in which parts not properly protected arrived at Port Elizabeth, South Africa, as a result of exposure to sea water.

Rustproofing and Packaging

Mr. Harper said that for the past several months the matter of rust-proofing and packaging of replacement parts for export has been the subject of considerable research by his company. "Under normal circumstances," he said, "when an export order had been packed it was railed to seaboard and in very few instances would it remain there for more than a month before being loaded into the hold of a boat to be carried to its final destination. On arrival the material would be immediately unpacked and placed in production or service stock." Today, he continued, because of wartime conditions cases must remain in shipyards or on railway sidings exposed to the elements for from four to six months awaiting a boat. After arrival at its destination, the material may remain in the cases for several months before being unpacked and made ready for use.

He expressed his company's appreciation for the cooperation given them by the Chrysler Corp. They have also been working closely with the Chrysler Corp. in Canada in the development of a standard procedure of protection which they are now in the process of instituting in their plants.

The joint presentation of Messrs. Shelly and Harper was an eye-opener to the audience – the largest regular meeting in the history of the Section.

Lt.-Col. Colby Gives Off-the-Record Address

▪ *Washington*

On Nov. 10 Lt.-Col. Joseph M. Colby gave an off-the-record talk to the SAE Washington Section on the subject, "Experience with Automotive Equipment in the Field." Lt.-Col. Colby, who recently returned from a tour of duty in North Africa, is chief of the Development and Engineering Branch, Tank-Automotive Center, Ordnance Department, Detroit. It was a capacity meeting and scores were turned away.

New England Section

Civilian Wartime MOTOR VEHICLE PROBLEMS

DEAN A. FALES

Associate Professor Automobile Engineering
Massachusetts Institute of Technology

IN the 10-year period from 1932 to 1942, performance and style of cars were stressed as sales features, but little thought was given to true economy of operation. The result was cars which were easy on the eyes, but in many cases harder and more expensive to service than the earlier and less attractive models.

Our present motor vehicles are vital to the life of the nation, and their preservation and maintenance are of utmost importance. Car owners who bought new models each year are going to discover that a properly maintained automobile can give many years of useful service – a fact that the larger number of used car buyers have known and profited by. In the past the engineers and stylists who created the models were the stars of the industry. The service men who kept them going – thereby creating repeat sales – were the unsung heroes of the industry. Now that we are in an all-out war, the maintenance men are the key men of civilian motor transportation.

Owner Must Share Responsibility

In order to adapt our vehicles to wartime operation the car owner must share the responsibility of caring for the car with the maintenance man. Although maintenance and service men are having their wartime problems of labor, parts, and material shortages, the owner must depend upon the service station to keep his vehicle in its best operating condition. Factory service organizations inform local stations of changes or alterations for improved economy and low-speed operation, and although some owners and service stations find ways of improving performance, all ideas should go through factory clearing houses, and nothing should be publicly recommended until it has been tried and approved.

A leading feature writer in the daily press championed the scrapping of all automobile bumpers, and many patriotic but misguided car owners stripped off their bumpers and gave them to the scrap drive. Bumpers should not be removed from automobiles in commission in the interests of highway safety and national economy. They contribute to the rigidity and strength of frames by acting as cross members. Bumpers are needed to jack up wheels on many cars; bumpers are the logical place to attach dim-out lights where such are required; bumpers are needed to push and start cars whose engines have stalled and which have no provision for hand starting; bumpers are the most convenient parts by which to tow or push disabled vehicles; bumpers protect the rear mounted gasoline tanks in minor rear-end collisions, thus reducing serious fire hazards; and bumpers protect otherwise vulnerable lights and their proper focusing.

If a city were to be evacuated suddenly, it would be largely done by motor vehicles, and traffic would be dense. If all cars had bumpers, and one car in the line faltered, it could be pushed or towed along or pushed out of the traffic line. But if a car or cars in the line-up had no bumpers, serious and dangerous holdups could occur.

As fixed charges remain the same, taxes increase, and mileage is curtailed, the cost per mile of operation may influence some owners to place their cars in storage. In order that their usefulness and value be preserved, cars should be stored properly. Maintenance men are qualified to prepare vehicles for storage, while most owners have only vague ideas about what should be done.

De-tuning cars from a snappy 90 to 100 miles an hour speed to a smooth and economical 30 to 35 miles an hour speed may call for minor mechanical alterations. Some owners may be able to change spark plugs, alter ignition timing, tinker with carburetion, fuels and lubricants. In the long run the more satisfactory and efficient method is to have this type of work done by men who are familiar with service work and who have had factory experience.

If the labor and materials shortage situation did not exist, new intake systems could be put on existing engines that would give economy at the expense of performance, but such major mechanical changes are out of the question now. Some successful changes which are being made are: Resetting ignition timing to care for changes in fuel; using hotter spark plugs for lower speed operation; using economy jets in carburetors, carburetor throttle opening stops or stronger springs on throttle controls; use of slightly higher than previously recommended pressures in tires, and so on.

Instructions Needed

For drivers who like to tinker with motor vehicles, there should be simple and clear instructions on wartime driving. For example, the accelerating pump on the carburetor aids snappy getaways, but it wastes fuel; if the accelerating pump is disconnected, performance will suffer, but fuel economy will gain. In some cases, the accelerating pump is used to assist in the starting of cold motors; if the accelerating pump is disconnected, cold starting may be difficult. If the driver understands the function and operation of the accelerating pump and will operate his foot throttle with a feather touch, he can have both economy and the original performance for starting and emergency acceleration.

The problems of the bus and truck operators are more serious in many respects than those of private passenger car owners. Buses and trucks have been operated under nearly full throttle conditions, and now they must operate under part throttle on the level, cannot rush hills (which will mean wide open throttles at low speeds on up grades), and cannot run down hills at increasing speeds (which may cause oil pumping). Many truck operators believe that 35 mph is too low a speed for economical and efficient operation with existing equipment, depot locations, and limited hours of continuous driving. The truck operator associations and Government officials will decide the issue of proper speed limits. Fortunately, bus and truck fleets are operated by experts who are qualified to make the changes needed in their equipment to cope with the new conditions.

The biggest and most important single factor in economical operation is the driver. We are a nation of automobile operators, and our skill has been developed along the lines of fast getaways and getting from one point to another as fast as traffic and the law allowed. We willingly paid the price of such operation by the use of excessive amounts of rubber, fuel, and oil. Now it is mandatory that drivers be schooled to a new and unfamiliar method of operation in order to conserve rubber, fuel, oil, and the vehicle itself. Unfit drivers as well as vehicles should and must be removed from the road. Needless and preventable accidents are as damaging and costly as enemy sabotage. Now is the time to use restraint and sound judgment in motoring.

[Mr. Fales presented his annual talk to the New England Section of the SAE at the Nov. 12 meeting of the Section.]

Gasoline Economies Despite Curtailments

Southern California

Some of the fuel and lubrication problems that the war has foisted on private and industrial vehicle users were discussed at the Southern California Section meeting held Oct. 30, at the Hollywood Roosevelt Hotel, Los Angeles. Gerthal French, Richfield Oil Corp., acted as leader of the discussion for the Lubricant Forum.

Mr. French based his problems on various Government orders requiring the reduction in SAE grades of petroleum products, recommending longer intervals between crankcase drains, and eliminating the use of steel drums and even some wooden containers for the delivery of petroleum products. Mr. Erickson, Adohr Milk Farms, said that his company uses only SAE 20 and 30 oil in its engines. Further discussion indicated that for cars and trucks these two grades would be sufficient; although R. E. Rowley, Department of Water & Power, Los Angeles, suggested that heavy-duty slow-moving equipment requires a heavier grade, SAE 40 or 50, to be added to the list. For hypoid axles, it was agreed that SAE 90 would probably be sufficient for most types of vehicles.

Speaking on the subject of "Octane Numbers of Gasoline and the Effects Resultant Therefrom," W. A. Moors, The Texas Co., said that although the octane number had already been reduced two points due to the Government order curtailing lead, the average driver probably hasn't noticed the difference. He mentioned that refiners were doing more cracking and reforming heavy ends to compensate for the loss of lead. Mr. Moors emphasized that if good driving habits are observed, lighter lubricants used, tires always kept fully inflated, carburetor adjusted properly, spark set for maximum economy, spark plugs kept clean and adjusted, and the air cleaner serviced regularly, surprisingly good gasoline economies still can be obtained.

The final speaker was William Dudley, Office of Price Administration, who explained how the nationwide gasoline ration, shortly to be put into effect, would work. He also tried to clear up the reason why gasoline rationing was necessary when there is so much gasoline in California. Since over 90% of the world's supply of rubber is in the hands of the Japanese, it is necessary for us to use every means possible to conserve our dwindling supply – and the greater portion of our supply is on the rims of wheels. Control of gasoline seemed to be the only solution.

Meeting Held at Parks Air College

St. Louis

The Nov. 3 meeting of the St. Louis Section was held at Parks Air College, East St. Louis, Ill. Charles G. Russell, inspection department of Parks Air College, spoke on the "Magnaflux Method of Inspection," and George Harsh, engine overhaul department of the college, presented a talk on the "Use of a Metallizing Gun." The meeting then adjourned to the shops where Messrs. Russell and Harsh demonstrated this equipment. Several members of the Parks Air College staff served as guides. Various engines including the Allison and Packard-built Rolls Royce were available for inspection.

Preparation Important In Metallizing Process

College of the City of New York

"Metallizing, Its Technique and Applications" was the subject of John Wakefield's talk at the Oct. 21 session of the SAE Club of CCNY. Mr. Wakefield, who is with the Metallizing Engineering Co., Inc., illustrated his subject with slides devoted to the reclamation of worn parts by building them up with sprayed metal. "Preparation is the biggest single factor in the success of any metallizing application, and its importance cannot be overstressed," he pointed out, adding, "select the method for the work to be done, and where a choice exists, use the method which produces the strongest bond.

"To begin with, the surface must be absolutely clean, free from paint, grease or oil, rust, scale, and all foreign matter," Mr. Wakefield continued. "Second, it must be roughened in such a way that anchors, hooks, and tears are provided – so that the particles driven against it have something into which to key themselves."

Mr. Wakefield also pointed out that, due to the present metal situation, the process not only saves the expense of a new part, but in many cases saves a part which could not be replaced if discarded.

General Motors Films Shown at Meeting

SAE Club of Colorado

Through the courtesy of the General Motors Corp., two films were shown at the Oct. 27 meeting of the SAE Club of Colorado. These were entitled "Diesel, The Modern Power," and "Precisely So," the latter about precision instruments.

Merits of the Aircooled In-Line Engine Discussed

College of the City of New York

The in-line aircooled engine was the subject of A. T. Gregory's talk at the Oct. 28 meeting. Mr. Gregory, chief engineer of Ranger Aircraft, expressed the belief that the aircooled in-line engine is the answer to the radial in-line controversy, as it incorporates the best features of both types.

Investigation has shown that on a pounds per horsepower basis the aircooled in-line is comparable to both other types, he revealed. In fact, the crankshaft assembly on the Ranger engine is lighter than that of an equivalent radial engine, in spite of its greater length. At present the upper limit of the Ranger engine is 500 hp, he continued, and it is expected that future development will bring it into the higher horsepower brackets.

Year's Program Plans Arranged

Oregon State College

The Oregon State College Student Branch formulated plans for the coming year at their Oct. 15 meeting. They intend to have two meetings with the SAE Portland Section.

Men in the aeronautical and automotive fields have been contacted for short talks, and motion pictures will be shown at the meetings whenever possible.

Chairmen Elections

At the Nov. 4 meeting of the Student Branch of the Oklahoma A and M College, the following committee chairmen were elected: Membership Committee, Eugene C. Bright; Program Committee, Bob Schuetz; Publicity Committee and SAE Field Editor, Dale Hugley.

Invitation to SAE Members

The New Zealand Division of the Institution of Automotive Engineers, Australia, extends a cordial invitation to SAE members visiting that country to attend meetings of the Institution, or take part in any of their activities.

D. P. Latimer, honorary secretary of the Institution, asks that correspondence be addressed to him in care of Box 1063, Wellington, N. Z.

Army Ordnance Decentralized In Wide Reorganization

RICHLY prophetic was the long-held opinion of the Army's new Chief of Ordnance, Major-Gen. Levin H. Campbell, Jr., who saw global warfare in the offing long ago. He is an engineer's engineer, and at the end of World War I was developing self-propelled mounts for cannon. Later he served under Pliny Holt, of pioneering tractor fame, and became chief of Aberdeen Proving Ground. Then at Rock Island Arsenal he became a manufacturing executive of major proportions, and has demonstrated a high ingenuity for applying mass production to armament products.

A measure of the General's capacity was his administration of the chemicals, explosives, and ammunition loading program of the Department, a vast plant totaling \$3½ billion.

It gives neither aid nor comfort to the enemy to point out that he picks his men carefully, assigns them jobs, and backs them and their judgment to the limit.

Key to General Campbell's reorganization – which began in July – was: DECENTRALIZED OPERATION.

This he has accomplished by:

1. Moving the top commands of seven important units from Washington out into the field, and

2. By increasing the authority and staffs of the Army Ordnance Districts.

The important administrative units moved out of the District of Columbia are:

• Security and Safety Branch, Chicago. Here in the center of the nation's industrial activity, Col. Francis H. Miles, Jr., and his staff undertake to reduce man-hour losses and increase manufacturing productivity by reducing accidents and controlling acts of sabotage – whether they be willful or the result of neglect.

• Tank - Automotive Center, Detroit. Brig.-Gen. Alfred R. Glancy, former General Motors executive, now Deputy Chief of Ordnance for Automotive – his new title. This is an example of how General Campbell gets things done. He picked a man in whom he has confidence, and then backed him to the limit. (See detailed organization of the T-A Center, next page.) Here General Glancy and his staff work *with* – not *at* – engineers who have assumed the responsibility of building the best automotive equipment possible for our Army.

• Field Director of Ammunition Plants, St. Louis. The General put this basic task in the capable hands of Col. T. C. Gerber and his staff of ammunition manufacturing, loading, and storage experts. Here is supervised that new \$3½ billion network of plants which General Campbell himself administered from a project to an entity.

• Small Arms Ammunition Sub-Office, Philadelphia. Historically, the Eastern Seaboard has been the nation's supplier of ammunition. Here, since the Civil War days, has been the home of developments of many major improvements in small arms ammunition, and Col. Boone Gross is in charge of this office. He is considered to be the country's foremost expert in this area of engineering and production. Frankford Arsenal, Philadelphia, has been the center of these developments.

• Inspection Gage Sub-Office, Philadelphia. Most of this work is in connection with small arms and small-arms ammunition, and this office is in the industrial center of this type of manufacturing.

• Field Service Maintenance Sub-Office, and the all-important Fire Control and Antiaircraft Unit are at Frankford Arsenal, Philadelphia. Major John A. Plimpon is in charge of these developments.

Main Divisions

Young, hard-hitting generals are in charge of the four divisions of Army Ordnance. These Division chiefs report directly to General Campbell, as does the Tank-Automotive Center's commander, General Glancy.

The four divisions of Army Ordnance are:

1. Military Training Division, under Brig.-Gen. Julian S. Hatcher. A graduate of the U. S. Naval Academy, he joined the Coast Artillery of the Army, and rose to lieutenant colonel by the end of World War I. Reverting to his permanent rank of captain after the Armistice, he came up the hard way to his present post. A scholar of world renown, he was picked by General Campbell to manage the far-flung training program of Army Ordnance – to teach the art of Ordnance handling and supply to the American Army. An expert in infantry and aircraft armament, he was appointed commandant of the Armament School in February of last year.

2. Technical Division under Brig.-Gen. G. M. Barnes. Another lieutenant colonel of World War I, General Barnes graduated from the University of Michigan in civil engineering, and also joined the Coast Artillery Corps. Reverting to his permanent rank of captain after the war, he served at Watertown and Frankford Arsenals, went overseas to be Assistant Ordnance Officer at Coblenz, Germany, and served in the post-war era in Rome. An authority on ordnance design, he has contributed widely to the development of railroad and automotive ordnance. He is today in charge of design and research, and is particularly close to the work of numerous SAE committees doing advisory work at his request. A number of SAE members, both in the Army and as civilians, report to the General on a wide field of ordnance development work. Results of his projects are then taken up by the

3. Industrial Division, headed by Major-Gen. Thomas R. Hayes, which contracts for weapons for our Army. He administers a \$52 billion business, works closely with automotive manufacturers and other metal-working industries on thousands of items needed by our armies all over the world.

The decentralized offices of the Ordnance District Chiefs report to General Hayes. There are 13 districts, and a total of 76 sub-offices which keep close liaison with General Hayes in the important task of integrating ordnance needs with the manufacturing facilities of the country. Each is in effect a miniature Army Ordnance Department.

Ordnance District offices are at:

Birmingham, under Col. Ernest C. Bomar, with two sub-offices;

Boston, in charge of Brig.-Gen. Burton O. Lewis, an SAE member;

Chicago, under Brig.-Gen. Thomas S. Hammond, who is in charge of eight sub-offices;

Cincinnati, which supervises nine sub-offices, under the direction of Col. Fred. A. McMahon;

Cleveland, under Col. Harold M. Reedall;

Detroit, Brig.-Gen. A. B. Quinton, Jr., in charge of seven sub-offices in Michigan and Canadian manufacturing centers;

New York, under Col. Robert W. Johnson, with a sub-office in Newark;

Philadelphia, with eight sub-offices, under the command of Col. David N. Hauseman, who has chosen SAE Member Charles O. Guernsey as his second in command;

Pittsburgh, under Brig.-Gen. Hugh C. Minton, with his 18 sub-offices;

Rochester, with six sub-offices under Col. Frank J. Atwood;

St. Louis, with six sub-offices under Col. Merle H. Davis;

San Francisco, in charge of Kenneth B. Harmon with three sub-offices, and

Springfield, Mass., the birthplace of American Ordnance development under Col. Guy H. Drewry, who has eight sub-offices reporting to him.

Thus the needs of the Army are brought closely to the manufacturing centers of the country. This is not a new policy, but one that has been extensively implemented by General Campbell.

4. Field Service Division, headed by Brig.-Gen. H. R. Kutz. This division is in charge of maintenance of all Ordnance equipment, and supply of weapons to the fields of operation. Formerly, the major supply function of

U. S. Army Ordnance Department

the Army was assumed by the Quartermasters Corps.

The chief of the Maintenance Branch of the Field Service Division is Brig.-Gen. James Kirk, and he is held responsible for the maintenance of weapons for our armies all over the world. Trucks, tanks, tractors, machine-gun carriers, and other motorized equipment – except aircraft – are his responsibility.

Tank-Automotive Center

The most radical departure made by General Campbell was the organization of the Tank-Automotive Center, referred to above, under Brig.-Gen. Glancy, as deputy chief of Ordnance for Automotive. He put this in the world's automotive center – Detroit.

CHIEF OF ORDNANCE

Maj.-Gen. LEVIN H. CAMPBELL, JR., responsible to Lt.-Gen. BREHON B. SOMERVELL for the design, manufacture, and servicing of Army Ordnance, and training of Ordnance personnel

Clelland his assistant chief, and SAE member Charles W. Kynoch, formerly a Chrysler engineering executive, the chief engineer of the Branch.

This branch has five engineering sections: Combat Vehicle, Motor Transport, Operations, Components, and Special Projects. They in turn have a total of 19 units, where the Development Branch engineering is done by military and civilian engineers.

Section Chiefs are:

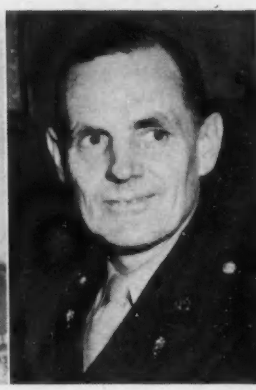
Major W. W. Durding, Combat Vehicle Section. His unit chiefs are O. J. Kangas, head of the Tank Unit. Capt. R. A. Fox is head of the Miscellaneous Vehicles Section. Capt. A. A. Parquette heads the Armored Car Unit, and Capt. J. O. Van Natta heads the Self-propelled Mount Unit.

The Motor Transport Section of the Development Branch is headed by Major E. M. Holtzkemper. His three units, with their chiefs, are Truck Unit, headed by Capt. A. E. Cleveland; the chief of the Trailer Unit is G. R. Engler; and the Head of the Motorcycle Unit is R. W. Enos.

TRAINING

Brig.-Gen. JULIAN S. HATCHER, in charge of the Military Training Division

ENGINEERING



Brig.-Gen. G. M. BARNES, in charge of the Technical Division of Ordnance

TANK-AUTOMOTIVE

Brig.-Gen. A. R. GLANCY, deputy chief of Ordnance for Automotive

MANUFACTURING

Maj.-Gen. THOMAS J. HAYES, in charge of the Ordnance Industrial Division

FIELD SERVICE

U. S. Army photos

Brig.-Gen. H. R. KUTZ, head of the maintenance and supply functions of Ordnance

He headed it with a civilian with wide executive automotive experience whom the Army commissioned a brigadier general, and he put Brig.-Gen. Donald Armstrong in as Chief of the Tank-Automotive Center.

Upon graduation from Columbia University in 1910, General Armstrong joined the Coast Artillery Corps, served in France with the AEF, and was assigned to the U. S. Embassy in Paris following the war. He became an Ordnance officer in 1923, studied the subject in France and this country, and has become a procurement and maintenance expert.

Assistant Chief of the Center is SAE Member Brig.-Gen. John K. Christmas, graduate in M.E. from Lafayette College in 1917, who went to France with the AEF early in 1918. A physicist and engineer of renown, he was transferred to the automotive division, Aberdeen Proving Ground, after a postgraduate course at Massachusetts Institute of Technology. He became chief of the automotive section, Artillery Division, in 1930, and has been concentrating his whole attention on tanks and combat vehicles since. He was chief of the former Tank and Combat Vehicle Section, Ordnance Department, and worked out the details of the first tank manufacturing programs with SAE Member Lt.-Gen. William S. Knudsen when the latter was in charge of the Office of Production Management. Col. E. S. Van Deusen is in charge of Engineering Liaison.

Brig.-Gen. DONALD ARMSTRONG (left) the Chief of the Tank-Automotive Center, Detroit, and Brig.-Gen. JOHN K. CHRISTMAS, assistant chief of the T-A Center

The Center, organized along the pattern of the Ordnance Department itself, has four main branches, each manned by experienced Ordnance officers and automotive engineers with long design and manufacturing experience. They are:

- Development Branch, headed by Lt.-Col. J. M. Colby, with Lt.-Col. C. K. Mc-

The Operations Section is headed by Capt. H. N. Brownson, an SAE member, formerly with Olds Motor Works. He has four units under his command. They are: Design Unit, headed by C. A. Rasmussen, another SAE member; Model Unit, in charge of Capt. F. W. Wayne, and an Engineering Test and a Proving Ground Unit.

The Components Section of the Branch is headed by Major D. N. Klima, who is in charge of developing improved component parts of tanks and combat vehicles. This work is divided among these six units: Capt. E. W. Swift, Armor and Armament Unit; Capt. D. O. Nichols, Engines and Power Trains Units; Capt. J. S. Rhyne, Electrical Unit; Capt. T. J. Hollenkamp, Running Gear Unit; and Capt. A. S. Avakian, Equipment Unit.

The Special Projects Section is composed of two units. One is the Rubber Conservation Unit headed by SAE Member Lt.-Col. B. J. Lemon, on leave from the U. S. Rubber Co., and who was transferred from the Quartermaster Corps in which he has been a reserve officer since World War I. Special Engine Project Unit is headed by Lt.-Col. A. B. Domonoske, an SAE member.

- Manufacturing Branch, under Col. M. E. Wilson. This Branch takes the designs

turn to page 42

CMP Is Born

Controlled Materials Plan Is WPB's New Hope

MOST realistic of World War II plans for controlling the flow of materials to armament factories is the new Controlled Materials Plan, announced to a plan-drunk press conference early last month by WPB Chairman Donald M. Nelson and an august assembly of top WPBers.

CMP provides:

- Seven agencies to get all scarce materials available. They are the Army, Navy, Maritime Commission, Lend-Lease, Board of Economic Warfare, WPB's Aircraft Scheduling Unit under Charles E. Wilson, and Civilian Supply;
- Each of these agencies will make up estimates for quarterly needs;
- These will be pared by negotiation if the requirements are beyond hope of attainment;
- When quotas for materials are approved, each agency will divide its allotment among its prime contractors. (Double check: Prime contractors may go over their requirements with WPB's materials branches;)
- Prime contractors will authorize subcontractors to buy material from suppliers.

Thus, for the first time, vertical allocations are the order of the day in so far as large tonnages of materials are concerned. The Production Requirements Plan, or PRP, or Purp, will be used to allocate materials for small items and shelf goods, such as bolts, nuts, washers, and smaller standard parts. Components and the end product will be assured of equal preference in materials.

"This is the closest approach by far to automotive shop materials scheduling," Ernest Kanzler, director general for WPB's Industry Operations, told the *SAE Journal*.

"The program is not perfect. A great deal of paper work must go out the window. But it is a start in the right direction, and shows a meeting of minds that is most important," he continued.

Apparent advantages of CMP:

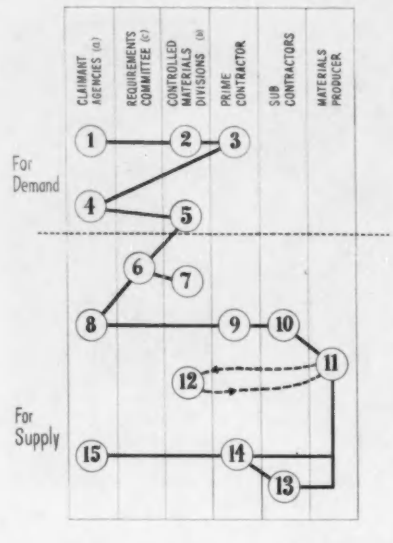
- Materials have finally got on a controlled scheduling plan that approaches industry's peacetime operations;
- For the first time, the procurement agencies will be forced to keep their requirements within the available supply of raw materials and will schedule output in terms of quarterly periods. If they fail, the responsibility and consequent criticism will be on the military, and
- The plan's fundamental framework is that of the Steel Budget Plan, and the end-product producers may continue the peacetime practice of controlling steel, non-ferrous, and materials scheduling at the mills, some WPB realists hope.

Apparent disadvantages of CMP:

- The oppressive slough of paper work continues, but there is hope in some industry and Government quarters that much of this can be eliminated;
- Processing of bills of materials is bound to be slow, may never get down to peacetime efficiency, and
- The plan itself is wide open to variant interpretations by WPB interpreters.

The fiasco of PRP with its PD25A forms, has brought both WPB and the military around to realizing that the war effort can't stand many more experiments.

Controlled Materials Plan, WPB
For Class A Manufactured Products



1. Claimant Agency (a) sets up a preliminary program for the manufacture of armament or military equipment, making its own estimate of the amounts of critical materials required. Often the Claimant Agency checks with
2. Controlled Materials Divisions (b), and
3. Contractor and such Subcontractors as have had specific experience in manufacturing the required item.
4. Claimant Agency then submits its demand (bills of materials multiplied by the prospective production schedules) to the
5. Controlled Materials Division, which gives it a preliminary review, and then notifies the
6. Requirements Committee (c), which totals the requirements of all Claimant Agencies, balances the demands against estimated available supplies, and makes adjustments. Thus the Requirements Committee has power to over-ride the Claimant Agencies on controlled materials. The Requirements Committee then notifies the
7. Controlled Materials Division of its decision.
8. Claimant Agency authorizes the
9. Prime Contractor an Allotment Number (d), and Preference Ratings (e). The Prime Contractor then divides the specific Quantity Allotments among his own plants and its
10. Subcontractors, and extends to his subcontractors the Allotment Number and Preference Rating which he received from the Claimant Agency. The Prime Contractor and his Subcontractors then order materials, with the Allotment Number, from the
11. Materials Producer, who reports these materials orders to the interested
12. Controlled Materials Division, which may issue Production Directives (f) to
11. Controlled Materials Producer, which ships the materials to the
13. Subcontractor and
14. Prime Contractor, who build the equipment and ship the product to the
15. Claimant Agency, thus completing the cycle.

(a) Claimant Agencies, first column.

(b) Controlled Materials Divisions: The former Iron & Steel, Aluminum and Copper Branches of WPB.

(c) Requirements Committee, WPB, is headed by Ferdinand Eberstadt, attorney and former securities broker, recently chairman of the Army-Navy Munitions Board, and now a vice-chairman of WPB. Serving as his chief lieutenant is Ernest Kanzler. Members are representatives of the Claimant Agencies (a, above), and the State Department.

(d) Allotment Number, a serial number which may be extended.

(e) Preference Ratings will be granted for other than Controlled Materials, to be known as Class B products – loosely defined as "shelf goods," such as bolts, nuts, etc.

(f) Production Directives are to be issued by the Controlled Materials Divisions to steel mills, aluminum and copper producers.

Aircraft Reorganized

WPB Branch Ended, Committee Is Formed Under New Board

SIGNIFICANT has been the emerging of the Aircraft Branch from the War Production Board to an important entity. The reorganization:

- Aircraft Production Board, and
- Aircraft Production Committee.

To all aircraft, engine, propeller, accessories, parts manufacturers and materials suppliers, the Aircraft Production Committee, headed by Theodore P. Wright as chairman, is more interesting than the Board. Reason:

For the first time since the earliest days of the Office of Production Management will the engineers who know the score be in charge of aircraft manufacturing for our armed forces and our allies.

A former vice-president of Curtiss-Wright Corp., and an outstanding aeronautical engineer, Mr. Wright has been "insulated" from top command with non-industry liaison since the days he resigned his post and undertook to help the American aircraft industry help win the war.

The Aircraft Branch was the first end-product industry branch in the OPM setup. It stemmed from an advisory committee to the President, and another committee of the National Defense Advisory Commission. Around this group was built OPM, and its successor, WPB. For this group were the critical materials branches formed in the first place, although recent history has indicated that ranking executives of the defense effort have overlooked this point.

Mr. Wright's able technical staff will move over to the Pentagon Building, across the Potomac, adjoining offices of the Army and Navy officers in charge of aircraft engineering and procurement. This staff of 200 will be paid by the Army and Navy.

Charles E. Wilson, vice-chairman of WPB in charge of production, will head the Board. Other members are Major-Gen. Oliver P. Echols, Rear Admiral Ralph E. Davison, and Harold E. Talbott, financier. Merrill C. Meigs, Hearst publisher in Chicago, resigned Nov. 15.

The Production Control Committee, under Mr. Wright, consists of Brig.-Gen. Bennett E. Meyers, AAF, and Rear Admiral Ernest M. Pace, Navy Bureau of Aeronautics.

Housing Standards Issued

Sweeping restrictions on housing standards have been made by WPB to insure obtaining all value possible from materials and labor used in home construction. Copper is practically tabooed (M-9-c-4).

The directive includes both Government-financed and private construction, and was issued as "War Housing Construction Standards."

Prices Under Control

The Office of Price Administration issued Price Regulation No. 251 (Oct. 31, effective Nov. 5), putting under complete control the prices of building construction.

U. S. War Production

Highlights of Official Report On Progress in Arms Manufacture

WITH the daily rate of war expenditure recovering following a sharp decline in October, actual production continues spotty. September releases showed, for example:

AIRPLANES: Nearly on schedule. However, shift to heavier combat types. Total value for the month was up 10%, however, as compared with a 5% gain in August. PROPELLERS continue to be a serious brake on aircraft production prospects. ENGINE output is well ahead of schedule.

ORDNANCE: Overall production rose 7%, a disappointing figure. TANK output was up 3%, but guns for tanks were ahead of schedule. ANTI-AIRCRAFT GUN production was good. AMMUNITION continued spotty, with excellent results in some areas, disappointing in others.

NAVY AND ARMY VESSELS: Deliveries of major types was ahead of expectations, smaller craft showed up lower than anticipated. Total gain was 22% over August.

MERCHANT VESSELS: Construction gains were 10% in terms of ships, 34% in tonnage. This was about 10% more than was expected. September production equalled the U. S. total during 1941.

MACHINE TOOLS: September showed for the first time during the present emergency actual decreases in backlogs, with a gain of 2.4% in deliveries, or a total volume of $120,118,000 for the month – a new all-time high.

CONSTRUCTION: The peak was reached in August, the nation having caught up with the projected overall program several months ahead of expectations. Total of military construction, ammunitions plants, and machinery and equipment was $1½ billions. The 1942 program runs to about $14½ billions, and represents about one-fourth of all Government expenditures.

TOTAL WAR COST: During September, $5½ billions were spent on the U. S. war effort, including Lend-Lease and other aid to our allies. October rose to $5.7 billions with a decline in the daily rate. This compares with $1.8 billions for October, 1941, and $1⅓ billion in October, 1940.

ASC Merged With Army Officer Procurement

The Army is back in full control of its officer procurement program with the consolidation of the Army Specialist Corps with the established Army Officer Procurement system, headed by a Personnel Board under Gen. Malin Craig, former Army chief of staff. Dwight F. Davis, formerly director general of the ASC, will serve as an adviser to General Craig.

Where critical need exists for the services of an engineer, records of these men will be studied by the Board, which will make recommendations to the Secretary of War. In general, the *SAE Journal* was told, such procedure will start at the office of the Army officer requiring specialized engineers.

SAE Aids Army To Secure Maintenance Engineers

E. F. Lowe, SAE assistant general manager, in charge of SAE West Coast Office (at the right) with friend. Mr. Lowe supplied a large number of engineers for Army

At the request of the Army, more than 260 automotive engineers and maintenance repair men have been interviewed in the SAE West Coast Branch Office, Los Angeles, and half of them placed through civil service or commissioned into the Army, E. F. Lowe, assistant general manager, reports.

Army interviewers of the Supply Division, Fort Douglas, Utah, assigned to obtain the needed men in Los Angeles, San Francisco, Portland, Seattle, Spokane, and other cities, were contacted by the SAE West Coast Office to aid in securing other qualified men also. SAE transportation and maintenance members of various West Coast Sections were asked to submit names and qualifications of maintenance men to supply the Army needs. Some of the best men secured by the Army were obtained through the SAE, an officer reported.

An Army officer, speaking at a West Coast SAE meeting in Los Angeles, said: "The civilian advisers recommended by the Society of Automotive Engineers have saved our neck on our maintenance problems. I was skeptical at first, but they proved to be a godsend."

Raw Materials Board Reports

NINE months of operation of the Combined Raw Materials Board were reported guardedly by William L. Batt, SAE member and American Member, and Sir Clive Baillieu, British Member. Highlights:

- An accurate statistical picture of the overall estimated needs and supply has been made. This, of course, is restricted for use of the Government agencies of both nations.
- Pooling technical experience in conservation of materials is now under way. Already large quantities of the more critical materials have been saved by changed specifications.
- Revised purchasing policies have eliminated competition among the United Nations. The flow goes first to the nations which need the materials most for winning the war.
- Coordinated search for new supplies of raw materials. This has led to some dramatic expeditions to discover new sources.
- Long term and spot allocations to various United Nations have been successful.

Price Controls OK'd

A three-judge Federal Court sitting at Wichita, Kan., unanimously upheld the constitutionality of the Emergency Price Regulation Act, and Federal wartime regulation of rents, Leon Henderson announced.

Rubber Horizon

Coordinated Research Expected To Speed Up Man-Made Rubber

The more scientifically-minded WPB officials are confidently expecting that the crude rubber business of the world will be on the rocks by the war's end. Reason:

Coordinated research of development work and the huge facilities of the rubber, petroleum, chemical and automotive industries, they point out, should show quick results. Implications:

- The entire Straits Settlements and Dutch East Indian economies would be wrecked if man-made rubbers could be compounded from domestic raw materials, and
- Any characteristic wanted in the end-product could be attained.

Tough-minded William M. Jeffers, Rubber Director named by Donald M. Nelson following the Baruch-Conant-Compton rubber report, expects that this development work can be speeded.

He and his associates in WPB are keeping a close watch over the pooled talent and research equipment of interested industries, and are not prepared to condone any dallying.

Mr. Jeffers, a railroad executive by training, has visited every important rubber factory, and has crowded months of normal survey effort into days. "A human whirlwind" his associates called him in describing several of his trips to rubber centers. He has great respect for technologists.

WPB High Command: Production Executive Committee Is Named

NEWLY organized, the Production Executive Committee of the War Production Board, is headed by Charles E. Wilson, vice chairman of WPB, on loan from the presidency of General Electric Co.

From here on in, according to Donald M. Nelson, WPB's chairman, Mr. Wilson "will be the top production authority in the war program, and will exercise the powers of the Chairman of WPB seeing to it that the production programs are met."

PEC will meet twice weekly. It brings together the top procurement of the war effort. Its members:

Lt.-Gen. Brehon B. Somervell, Commanding General, Services of Supply, U. S. Army;

Major-Gen. Oliver P. Echols, Commanding General, Materiel Command, Army Air Forces;

Vice-Admiral Samuel M. Robinson, Director of Materiel & Procurement, U. S. Navy, and

Rear-Admiral Howard L. Vickery, Vice Chairman, U. S. Maritime Commission.

Mr. Wilson has been president of G.E. since 1940.

U. S.-British

Exchange Missions Seek to Integrate Techniques, Ideas

Recent developments in the area of war missions have been reported by the War Production Board. Others, chiefly military, are seldom announced.

• Following a British mission tour of American factories recently, eight top production executives of the Martin, Boeing, Consolidated and other plants have returned. They were under the direction of Theodore P. Wright, SAE member and deputy director, WPB's Aircraft Production Division, and now chairman of the new Aircraft Production Committee.

• A function of another mission, about to leave, is to study and plan closer correlation of American and British standardization.

• A British mission is studying the American materials allocation program, now the Controlled Materials Plan of WPB. An American mission has been studying the British method.

• Mica mission, in this country.

Capacity Upped

Within a year the $26 million addition to the Kaiser Co., Inc., steel plant at Fontana, Calif., will be in production. This brings the West Coast development to a total of 657,000 tons of ingot annually. The first part of the program will be in production during the first quarter of next year, with an estimated capacity of 450,000 tons of ingot a year, or 300,000 tons of plate.

Steel Alloys

DOMESTIC consumption of alloy steels during 1942 will double that of 1940, WPB estimates.

How?

• By reducing the alloying content of steels for armament production, through the National Emergency (NE) steels. Says WPB: "In conjunction with the industry and with technical groups, WPB has speeded the combination and substitution of the more plentiful alloying elements for the scarcer ones."

• By increasing production of the alloying materials, and

• By industry and Government expansion of alloying capacities of steel mills.

Alloying material capacity has been upped thus:

• **Chromium:** Domestic output in 1943 will be as large as prewar imports, or about 250 times our prewar production. Low grade ores are being developed in Montana, California and Oregon, principally. Cost to Government will be about $10 million and to private industry $2¼ million. With reduced imports and stock-piles, there won't be enough, however, except for barest military necessity.

• **Molybdenum:** U. S. production ran ahead of consumption during peacetimes, and considerable stockpiling was resorted to by the Government. However, moly was used to conserve other critical materials, and deep cuts have been made in stocks. The 1943 production will be upped by 15% over 1942, with old and new deposits being developed.

• **Tungsten:** The domestic production of 3600 tons in 1939 was doubled in 1941, and by the end of this year will again be doubled to an estimated 15,000 tons. Rich veins of the metal were recently discovered in Idaho. Despite these facts, 1942 shortages will be about 10%.

• **Vanadium:** U. S. 1942 production will double the 1937 figure this year, largely through Government financing of low grade ores. Peru is the largest producer, the U. S. second. In 1943, this will again be doubled, WPB estimates.

• **Nickel:** War needs for 1942 will total four times the domestic consumption of 1938. The 1943 production may double this year's. Cost: $100 million being spent by industry in Canada, $20 million by the Government in Cuba, and about $10 million by industry and the Government on the few known small U. S. deposits.

• **Manganese:** Prewar imports constituted most of the U. S. consumption. About 600,000 tons will be produced in 1943 in the U. S., against the 30,000 tons produced here in 1939. Cost to the Government is about $40 million, to private industry $6 million.

Ford Plant Sold

Negotiations by WPB resulted in the sale by Ford Motor Co. to the Russian Government of its complete tire manufacturing plant and other movable facilities, new Rubber Director William M. Jeffers announced.

Ordnance Reorganization

continued from page 39

developed by the Development Branch, and sees that the tanks and combat vehicles are manufactured. Under Col. Wilson are four sections. They are:

Engineering Section, headed by Lt.-Col. E. L. Cummings; and Capt. H. G. English heads the Product Appraisal Unit.

Lt.-Col. F. R. Young is chief of the Materiel Engineering Unit, which is the focal point of materials specifications. An Engineering Records Unit is headed by Lt. N. Druckleib.

The Washington Section is headed by Lt.-Col. J. B. McInerny, and his assistant is Henry Howard, formerly with General Motors Corp.

The Production Section is headed by Lt.-Col. George White. Colonel White also heads his own Production Control Unit, and Major Harry Haas is in charge of the Procurement Processing Unit.

The Inspection Section is headed by Lt.-Col. J. G. Hritz. His four units are Mail and Records, Industry and Field Inspection, Special Components, and Depot Inspection.

• Supply Branch, headed by Col. P. G. Rutten, is composed of four operating sections with a total of 14 units.

The Automotive Section is headed by Lt.-Col. W. K. Ghormley. He has three units: Control and Coordination, Vehicle Preparation, headed by Lt.-Col. J. G. Folger, and a Distribution Unit, of which Capt. C. J. Arcilesi is in charge.

The Tools and Equipment Section of the Supply Branch is headed by Lt.-Col. H. E. Hochberg. He has a Procurement Follow-up Unit, a Technical Unit, a Requirements Unit headed by Capt. C. R. Lathrop, and a Distribution Unit.

The Parts and Supply Section of the Branch is headed by Lt.-Col. M. E. Sheahan, and his assistant is Lt.-Col. J. A. Barclay. There are five units in this section. They are Technical Unit, headed by W. L. Wright, a Procurement Follow Unit, Emergency Procurement Unit, Requirements Unit under Major A. J. Plant, and a Distribution Unit under J. A. Burnett.

The Storage and Issue Section is headed by Lt.-Col. W. W. Townsend. His two units are Traffic and Space, headed by E. J. Hern, and Operations Unit, in charge of C. L. Daly.

• Maintenance Branch, which is headed by Col. S. E. Reimel, has five sections, which in turn divide their work among a total of eight units.

The Technical Section is headed by Major F. G. Pratt. His three units are: The Tank Unit, in charge of Capt. H. G. Ward, Wheeled Vehicle Unit, under Capt. O. M. Pease, and the Tractor Unit headed by Capt. J. H. Howard.

The Operations Section of the Maintenance Branch is headed by Major Harry O. Mathews, a former vice-president of the SAE, and a motor-vehicle operator of wide experionce. He has these three units under his command: Shop Unit, Capt. C. F. Venn; Modification Unit, headed by Major Val F. Gruenwald, another SAE member; and a Reclamation Unit, in the charge of Major A. F. Hoch.

The Parts Requirements Section of the Branch is headed by Lt.-Col. F. D. Rankins. His Operating Unit is headed by Capt. C. R. Vehrs and his Publications Unit is under Capt. G. D. Spies.

About SAE Members

ARCH L. FOSTER is a member of the *Oil and Gas Journal* staff, Tulsa, Okla. We stated in error in the November *SAE Journal* that Mr. Foster was with *National Petroleum News*.

O. E. **HUNT** has been elected an executive vice-president of the General Motors Corp. Mr. Hunt, an SAE member since 1915, has been a vice-president of the company since 1929, and is a member of the board of directors. He joined the Packard

O. E. Hunt

Motor Car Co. in Detroit in 1909, a position which he held until 1918, when he resigned to become associate district manager at Detroit and chief engineer in charge of Liberty aircraft engine design and production for the United States Government. Following World War I, Mr. Hunt re-joined Packard as chief engineer of the car division. Later he became vice-president in charge of operations for Hare's Motor in New York, and then he was chief engineer for the Chevrolet Motor Co. in Detroit.

Formerly in the purchasing department of Trico Products Corp., Buffalo, **JULIAN R. OISHEI** is now process engineer in the procurement engineering department.

GLEN F. JENKS has joined the staff of Taylor-Winfield Corp., Warren, Ohio, as consulting engineer.

WILLIAM A. CATALANO, who had been automotive engineer of the Standard Oil Co. of Ohio, is now field service engineer of the White Motor Co., Cleveland.

KENNETH T. MILNE has been named sales manager of the Delco Radio Division of General Motors, Kokomo, Ind., having previously been carburetor engineer.

F. EUGENE NEWBOLD, JR., is now assistant to the executive engineer of Ranger Aircraft Engines, Division of Fairchild Engine & Airplane Corp., Farmingdale, L. I., N. Y. He had been equipment engineer.

CHARLES J. SOSS has been made president of the Soss Mfg. Co., Detroit. He had been treasurer of the company.

ROBERT C. ENGELMAN, design engineer for the Clark Bros. Co., Olean, N. Y., received an award from the James F. Lincoln Arc Welding Foundation of Cleveland. His paper was an important unit in the Lincoln Foundation study on arc welding which covered every phase of industry.

DALE D. BALDRIDGE has joined the U. S. Forest Service, Department of Agriculture, in Patterson, Calif., as foreman of the Branch Shop.

"Wartime Chemicals from Natural Gas" was the subject of an address by **DR. GUSTAV EGLOFF,** president of the American Institute of Chemists, and director of research for the Universal Oil Products Co., at a recent testimonial dinner given in his honor. Dr. Egloff has been appointed a member of the Permanent Council for World Petroleum Congresses, and has written numerous books on petroleum chemistry. He now serves as member or adviser on five Government war planning and coordinating committees.

JOHN A. SPANOGLE is chief of the New Building Projects Unit, Power Plant Laboratory, U. S. Army Air Forces, Materiel Center, Wright Field, Dayton, Ohio.

Formerly automotive engineer for McKesson & Robbins, Inc., Fairfield, Conn., **EMORY FRANCIS PHELPS** is now a mechanic at J. L. Lucas & Son, located in the same city.

ERWIN BALLUDER has been transferred from manager of the Western Division of Pan American Airways, Brownsville, Tex., to the position of executive assistant to the vice-president, and is located in Pan American's New York office.

The Ahlberg Bearing Co., Chicago, recently announced the appointment of **CHARLES NELSON, JR.** to the position of chief engineer of the company, in charge of bearing design and development work. Mr. Nelson joined the Ahlberg staff in 1931, was made assistant chief engineer in 1933 and continued in that position until his present appointment.

LEONARD W. REEVES, formerly manager of Eastern Sales, Thompson Products, Inc., Cleveland, is now connected with the Toledo Steel Products Co. as vice-president and general manager.

WALDEMAR O. BISCHOFF is an air compressor designer at the Worthington Pump & Machinery Corp., Holyoke, Mass. He was formerly a designer at the American Bosch Corp. in Springfield, Mass.

SEWARD N. LAWSON is now an ordnance engineer, Detroit Ordnance District. Previously Mr. Lawson had been a partner in the Moore Laboratories, Detroit.

DALE L. COSPER has joined the Donnelly Engineering Co. of Detroit as aircraft process engineer. He had been full-size-body layout draftsman and assistant to the director of design at the International Harvester Co., Fort Wayne, Ind.

M. M. HENRY has been transferred from the Delco Remy Division of General Motors, Anderson, Ind., where he was engineer, to the Corporation's Eastern Aircraft Division at Linden, N. J., as armament engineer.

ROBERT M. BARNHART is purchase contact engineer for the Southern California Division of the General Motors Corp. at South Gate. Mr. Barnhart had been assistant engineer of the Utility Trailer Mfg. Co., Los Angeles.

Formerly test engineer at the Wright Aeronautical Corp., Paterson, N. J., **JOHN DeWITT, JR.** has been made an assistant project engineer.

CLYDE L. SAVAGE is now connected with Norman E. Miller & Associates, Inc., Detroit, as design engineer. He leaves the Standard Tube Co., of the same city, where he was chief engineer.

W. H. C. WHITE has been granted a leave of absence by the Sun Oil Co., Ltd., Toronto, for the duration and is now serving with the Department of National Defense at Ottawa. Mr. White has been with Sun Oil as an automotive engineer.

Formerly vice-president and general manager of Standard Products Co., Detroit, **W. C. IRELAND** has joined the Bundy Tubing Co., of the same city, as vice-president in charge of operations.

H. LIGGETT GRAY is donating through the Bureau of Navigation, a silver plaque as an award for achievement to the first vessel of the 110 ft P. C. class which distinguishes itself by outstanding action against the enemy. The plaque suitably inscribed will be in commemoration of the officers and men who lost their lives in the sinking of the S. C. 209 on Aug. 27, 1918. Mr. Gray is vice-president of Oakite Products, Inc., New York City.

COL. JOHN H. JOUETT resigned as president of the Aeronautical Chamber of Commerce to take charge of the aviation division of Higgins Industries, Inc., which builds Army transports. These transports

Col. John H. Jouett

will not be the regular C-46 Commandos, but an adaptation in plywood known as the C-76. **ANDREW J. HIGGINS,** president of the Higgins company, confirmed the fact that he had been offered a contract for 1200 big cargo planes and had accepted it. Mr. Higgins said that the planes would be constructed of non-strategic material, but would not say what type they would be, although he admitted they were not the radical multi-engined design he had had in mind. Unless unforeseen obstacles arise, he said, he will turn out the first plane within six months after getting the contract.

"The machine tool industry, after the war, will find its markets through new progress in machine design, through rehabilitation work in Axis nations as well as in the United Nations, and in the fact that we are heading for a socialized world when the United States must bring to the rest of the world the benefits of the social and economic life which we live," believes **E. PAYSON BLANCHARD**, sales manager of The Bullard Co.'s machine tool plant in Bridgeport, Conn. Mr. Blanchard expressed his opinions at a gathering of business men from Bridgeport.

EDWARD S. TAYLOR is professor of aircraft engines at the Massachusetts Institute of Technology, Cambridge. He had been associate professor of aeronautics.

GEORGE H. STOUGHTON, who had been a dynamometer test engineer of the Detroit Diesel Engine Division of General Motors, has been transferred to the position of experimental engineer, the same company.

ERNEST H. ALLEN has been assistant superintendent of the Outfitting Dock at the Albina Engine and Marine Works for several months. Mr. Allen was formerly owner and manager of the Allen Battery Co., Portland, Ore.

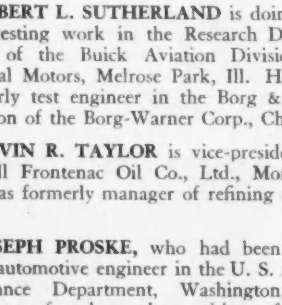
Formerly equipment and machinery designer of the Timm Aircraft Corp., Van Nuys, Calif., **NELS E. NYLIN** is now tool engineer of the Consolidated Aircraft Corp., San Diego, Calif.

WILLIAM B. HURLEY is on leave of absence from his position of staff engineer in the sales department of the Detroit Edison Co. He is at present working full time as assistant district chief of the Detroit Ordnance District.

ROBERT L. SUTHERLAND is doing engine testing work in the Research Department of the Buick Aviation Division of General Motors, Melrose Park, Ill. He was formerly test engineer in the Borg & Beck Division of the Borg-Warner Corp., Chicago.

GAVIN R. TAYLOR is vice-president of McColl Frontenac Oil Co., Ltd., Montreal. He was formerly manager of refining operations.

JOSEPH PROSKE, who had been principal automotive engineer in the U. S. Army, Ordnance Department, Washington, has been transferred to the position of chief



Joseph Proske

engineer, Engineering Section, Manufacturing Branch, U. S. Army, Ordnance Department, Tank-Automotive Center Engineering Office, in Detroit.

General mechanical engineer for the New York Air Brake Co., Watertown, N. Y., **A. W. LAIRD** has also become works manager of Hydraulic Controls, Inc., Chicago.

C. W. KIRKPATRICK has just returned from England where he spent the past two years on the technical staff of Canadian Military Headquarters. He is now experimental engineer in the Army Engineering Design Branch, Department of Munitions and Supply, Ottawa, Ont., Canada.

L. S. PFOST, SAE vice-president representing Tractor and Industrial Engineering, is now a member of the SAE War Engineering Board. Mr. Pfost is with the Massey-Harris Co., Racine, Wis.

A leave of absence has been granted **W. V. HANLEY,** research engineer of the Standard Oil Co. of Calif., Richmond. Mr. Hanley will become engineering test pilot in charge of power plant problems for the Intercontinental Division of TWA, Washington, and due to his new location he is resigning as treasurer of the SAE Northern California Section.

Addition of **ELMER P. GUTH** to the staff of field service engineers of the Pesco Division of the Borg-Warner Corp., Cleveland, has been announced by the company. Mr. Guth was formerly a hydraulic engineer for the Waco Aircraft Co., Troy, Ohio.

Dr. R. P. Dinsmore

Called to Washington to organize and coordinate the nation's efforts in the development of synthetic rubber, **DR. R. P. DINSMORE** was given an indefinite leave of absence from his position as research and development chief of the Goodyear Tire & Rubber Co.

G. N. GASCOIGNE has been promoted to the position of manager of fleet sales of the National Refining Co., Cleveland. He was formerly in the lubrication department. Mr. Gascoigne, a member of the SAE Transportation & Maintenance Committee, has been associated with the automotive industry for over 20 years.

WILLIAM LICHTY has enrolled in a production engineering course at the General Motors Institute in Flint, Mich. His cooperating unit is the Cadillac Motor Co., Detroit, where he will be employed during his work months in the engineering department.

BENJAMIN PFEIFFER has been transferred from the U. S. Navy Yard in Pearl Harbor, where he was material engineer, to the War Department, Chicago Ordnance District.

REX J. LEON DUTTERER, who had been project engineer in the engine development department of the Continental Aviation & Engineering Corp., Detroit, and consulting engineer for the Defiance Spark Plug Corp., Toledo, has joined the staff of Motor Master Products Corp., Chicago, as chief engineer.

At the 10th Annual Meeting of the National Lubricating Grease Institute, held in New Orleans in the latter part of October, **CHARLES B. KARNS** was elected president for the ensuing year. Mr. Karns is vice-

Charles B. Karns

president of Penola, Inc., Pittsburgh, and general manager of the Pittsburgh Plant of the Standard Oil Co. of Pennsylvania. He has represented the Standard Oil Co. of New Jersey and its many subsidiaries as a director of the National Lubricating Grease Institute for a number of years.

ERNEST R. STERNBERG is no longer business analyst for the WPB, Washington, having been changed to the position of assistant chief of the War Agencies Section, Automotive Branch, WPB.

W. D. ELLISON is supervisor of aviation service, General Petroleum Corp. of Calif., Los Angeles. Mr. Ellison had been senior commercial sales representative of the company.

Formerly chief technical engineer, Reece Button Hole Machine Co., Boston, Mass., **WILLIAM W. DUNNELL JR.** is now project engineer at the Massachusetts Institute of Technology, Division of Industrial Cooperation, Cambridge, Mass.

DON V. EELLS has been promoted from assistant plant manager to plant manager of the Cities Service Oil Co.'s refinery department, Ponca City, Okla.

Formerly chief draftsman of the Curtis Pump Co., Dayton, Ohio, **RICHARD LAWRENCE GATES** has been transferred to the position of project engineer.

HANS RUTISHAUSER has joined the Lycoming Division of The Aviation Corp., Williamsport, Pa. Mr. Rutishauser was formerly a designer in the Diesel Division of the American Locomotive Co., Auburn, N. Y.

CHARLES W. PHELPS has left the staff of Purdue University and is now on the mechanical engineering staff of the School of Engineering at Yale University, New Haven. For several months after leaving Purdue, Mr. Phelps did engineering work with the project engineers who are in charge of the Pratt & Whitney Double Wasp Engine.

T. P. WRIGHT, deputy director of the Aircraft Branch, WPB, had a lengthy audience with King George at Buckingham Palace recently, and told him about the mis-

sion to England of the U. S. aeronautical experts, which Mr. Wright heads. Mr. Wright and his colleagues visited plane factories in the northeast area in their study of British aircraft production methods.

ALEXANDER HAUSER is chairman of the Post-War Committee of the New York Board of Trade. Mr. Hauser is a director of the Standard Bending Co. and the Latin-American Credit Corp., New York, and is a member of the SAE's Membership Committee.

Alexander Hauser

Formerly assistant to the president on Government work, Firestone Tire & Rubber Co., Akron, Ohio, **RUSSELL ALLEN FIRESTONE** is now general manager of the Nebraska Defense Corp., Fremont, Neb.

J. J. MILLER, who had been automotive engineer, has been transferred to the position of senior lubrication consultant, General Petroleum Corp. of California, Los Angeles.

CHARLES D. HEALEY has been promoted from maintenance supervisor to supervisor of automotive transportation, Tide Water Associated Oil Co., New York City.

CALEB S. BRAGG has left the Langley Aviation Corp., New York City, where he was president, and is now vice-president of C. M. Keys Aircraft Service, Inc., also of New York City.

Formerly in the design and development department of the Hudson Motor Car Co., Detroit, **CARL W. CENZER** has been promoted to assistant project engineer.

FLOYD CLEVELAND KNIGHT has left his position of assistant professor of mechanical engineering, Case School of Applied Science, Cleveland, and is now chief engineer of Una Welding, Inc., the same city.

JAY A. YOUNG is no longer connected directly with the automotive industry. At present he is working as an ordnance engineer for the War Department, Office of Chief of Ordnance, Industrial Division, Ammunition Branch, Production Engineering Section, Bomb and Pyrotechnique Group.

NELSON G. KLING, formerly a designer in the Aircraft Division of Electrol, Inc., Kingston, N. Y., has become chief engineer of the Airex Mfg. Co., Long Island City, N. Y.

KINGSLAND HOBEIN has been transferred from Wright Aero, Ltd., Los Angeles, to the Wright Aeronautical Corp., Paterson, N. J. Mr. Hobein is a field engineer.

MARVIN G. HARRISON, formerly vibration engineer at the Curtiss-Wright Corp.'s Propeller Division in Clifton, N. J., is now mathematics consultant to an Aerodynamics Group at the Republic Aviation Corp., Farmingdale, L. I., N. Y.

PAUL G. HOFFMAN, president of the Studebaker Corp., was re-elected chairman of the United China Relief.

CHARLES O. BECH, who had been layout draftsman at the Rogers Diesel & Aircraft Corp., New York City, is now service engineer with the Farrel-Birmingham Co., Inc., New York City.

Formerly automotive technician, U. S. Army, Quartermaster Corps, Holabird Motor Base, Baltimore, Md., **GORDON G. ALLAN** is at present associate engineer in the Army Ordnance Department's Tank-Automotive Center, with headquarters in Detroit.

DONALD FAIRBAIRN, of the B. F. Goodrich Co., has been transferred from his position of sales engineer in the War Products Division, Akron, Ohio, to district manager of the California district, National Sales & Service Division of the company, Los Angeles.

ROY A. GARNETT severed his connection with the Whiting Corp. Ltd. (Canada), Toronto, where he was chemical engineer, and has become chemical and mechanical engineer of Sherbrooke Machineries, Ltd., Sherbrooke, Que., Canada.

The Cleveland Chamber of Commerce recently announced the appointment of **ALBERT J. WEATHERHEAD, JR.,** to the board of directors. Mr. Weatherhead is president of the Weatherhead Co., Cleveland.

CHARLES H. McCREA has been elected president of the National Malleable & Steel Castings Co., Cleveland. Mr. McCrea has been with National Malleable since 1913, his continuous association with the company being broken only by service as captain in the U. S. Army during World War I. In May, 1942, he was made first vice-president and a director of the company, holding this position until his present appointment.

Obituaries

John L. Gonard

John L. Gonard, retired inventor, died Nov. 1. Mr. Gonard, who was 75 years old, was the owner of patents for diamond cutting machines and a device for preventing retrograde movement of motor vehicles. He retired from his garage and repair shop in Englewood Cliffs, N. J., in 1924, and continued experimental work in his machine shop at home.

Ray F. McNamara

Ray F. McNamara, parts engineer at the Chrysler Corp., Detroit, died of a heart attack several weeks ago while returning to Detroit from a New York trip. Mr. McNamara, 56 years old, worked at the Premier Motor Mfg. Co. in Indianapolis for 12 years, in the service and sales department. From 1914 to 1925 he was with the Maxwell Motor Sales Corp., Detroit, handling contest work, publicity, and the activities of engineering service field representatives. He joined Chrysler in 1926.

Charles F. Magoffin

Charles F. Magoffin of the British Purchasing Commission, Washington, died recently at the age of 59. Mr. Magoffin was chief experimental engineer with the Pierce Arrow Motor Car Co. of Buffalo until 1922. He was engineer in charge of commercial and bus design at the Reo Motor Car Co., Lansing, Mich., for seven years. Long experience in the automobile industry well fitted him for his position with the British Ministry of Supply.

John J. Klein

John J. Klein, assistant to A. W. Herrington at the Marmon-Herrington Co., Inc., Indianapolis, died Nov. 11 of pneumonia. He was 32 years old. Mr. Klein, a graduate of the Fordham Preparatory School, the Stevens Institute of Technology, and the Fordham University School of Law, was admitted to practice before the bar in the State of New York, the State of Indiana, and the U. S. Supreme Court.

Explains New Process

COL. WILLARD F. ROCKWELL (right), board chairman, Timken-Detroit Axle Co. and director of production, U. S. Maritime Commission, explaining a new gear manufacturing process to Robert R. Nathan, chairman of War Production Board Planning Committee. The Timken-developed forge process will speed gear making and save millions of pounds of alloy a year.

In Military Services

MAJOR-GEN. JAMES H. DOOLITTLE, holder of numerous speed records as a civilian flier, and an engineering executive of the Shell Oil Co., Inc., advocate of the utmost in precise maintenance of equipment, is in charge of the U. S. bomber and fighter aircraft in the American "second front" in Africa.

SAE Member Heads United States Air Offensive in African Campaign

Formerly with the Bower Roller Bearing Co., Detroit, **R. H. DuBOIS** is now a lieutenant (j.g.), U. S. Navy.

MOORE KELLY, JR., has left the Boeing Aircraft Co., Seattle, where he was an inspector of finished material, and is now in the U. S. Army, Company D, Quartermaster Corps 203, Camp Young, Indio, Calif.

WILLIAM C. MORGAN, JR., has been promoted from major to lieutenant colonel, and is on overseas duty. Previously he was in the Office of the Quartermaster General, Motor Transport Division, Washington.

COL. P. H. ROBEY is in the Experimental Engineering Section, Power Plant Laboratory, U. S. Army Air Forces, Materiel Center, Wright Field, Dayton, Ohio. He was recently promoted from lieutenant colonel.

MAJOR PAUL FRANKLIN NAY, Army Air Forces, Materiel Center, Wright Field, Dayton, Ohio, is chief of the engine research sub-unit, power plant laboratory. Major Nay was promoted from the rank of captain.

CAPT. W. H. BOSHOFF, U. S. Army, Ordnance Department, has been transferred from Camp Blanding, Jacksonville, Fla., to Fort Bragg, N. C.

Formerly sales engineer, Continental Oil Co., Ponca City, Okla., **FRED B. HUNT** is now at the Aberdeen Proving Ground, Md., in Company A, Headquarters Ordnance Training Battalion.

LT.-COL. CARL E. CUMMINGS, Army Ordnance Department, has been transferred to the Tank and Combat Vehicle Center, Fisher Building, Detroit. Recently promoted from the rank of major, Lt.-Col. Cummings was supervisor of the research department, The Texas Co., Beacon, N. Y. He joined the Army in 1941 as reserve captain.

Formerly vice-president of engineering and maintenance, Pennsylvania Central Airlines Corp., Pittsburgh, Pa., **COL. LUTHER HARRIS** is chief engineering officer, Mobile Air Depot, Ala.

2ND LT. WALTER S. FORTNEY is in the Army Air Forces, Air Service Command, Air Depot Training Station, New Orleans, La. Lt. Fortney is a student engineering officer.

BRITTON L. GORDON has been promoted from lieutenant to major, and was transferred from the Chicago Ordnance District office to the Office of the Chief of Ordnance, District Administration Branch, Arlington, Va.

B. V. ROMERSI is in Company D, Third Battalion, Aberdeen Proving Ground, Md. Before entering the service, he was American representative of Fiat, S. A., Detroit.

2ND LT. NILE E. FAUST is an assistant engineering officer in the Army Air Forces, Rome Air Depot, Rome, N. Y.

CAPT. HAROLD N. BROWNSON has been transferred from the Main Post, Aberdeen Proving Ground, Md., to the Tank-Automotive Center, Development Branch, Detroit, Mich.

RUSSELL H. JOHNSON, who had been an automotive designer at the General Motors Corp., Detroit, is now a major in the U. S. Army, Ordnance Department, and is headquartered at Lehigh University, Bethlehem, Pa. Major Johnson is an assistant professor of military science and tactics.

2ND LT. ALBERT M. PATTERSON is assistant engineering officer in the Army Air Forces, Dale Mabry Sub Depot, Tallahassee, Fla.

GEORGE ROMNEY, managing director of the Automotive Council for War Production, delivered an address entitled "The Time for Talk Is Past," at the fall convention of the Michigan Commercial Secretaries Association. "The spirit of the automotive industry is, it's more important to help each other than to blame each other," he said. "Let's stop talking so much and do all we can."

The announcement recently appeared in the press of the appointment by President Roosevelt of **RALPH E. FLANDERS,** president of the Jones and Lamson Machine Co., to the Economic Stabilization Board, which will advise James F. Byrnes, Economic Stabilization Director, in the discharge of his duties.

W. K. ATCHESON, formerly superintendent of the S. W. Producing Division, Motor Transport Department, Pure Oil Co., Tulsa, Okla., is now a major in the U. S. Army, Engineer Amphibian Command, Camp Edwards, Mass.

COM. CLARENCE E. EKSTROM, USN, Naval Air Station, Corpus Christi, Tex., has been promoted from lieutenant commander.

ENSIGN WILLIAM C. MORRIS, USNR, is stationed at the Philadelphia Navy Yard. He was formerly assistant automotive engineer, War Department, Office of the Chief of Ordnance, Tank and Combat Vehicle Division, Washington.

W. S. JOHNSTON, Army Air Forces Resident Representative, Propeller Division, Curtiss-Wright Corp., Caldwell, N. J., has been promoted from major to lieutenant colonel.

VAL F. GRUENEWALD is now a major in the Ordnance Department of the U. S. Army. In civilian life Major Gruenewald was superintendent of motor transportation, Pure Oil Co., Atlanta, Ga.

MAJOR WILLIAM H. FISHER, formerly with the Gulf Oil Corp., has been relieved as instructor of engines at Fort Benning, Ga., and is now commandant of the Automotive School at Camp Croft, S. C.

Among SAE student members who recently entered the services are: **LT. ROBERT L. MAXWELL,** University of California, now a U. S. Army Air Reserve; **LT. CARL E. WULFF,** University of Wisconsin, Ordnance Maintenance Branch, 8th Armored Division, Fort Knox, Ky.; **FILO H. TURNER, JR.,** ensign, USNR, who is assigned to aviation specialist duties; **LT. SAMUEL P. ALTMAN,** College of the City of New York, now in the 206th Ordnance Company, 6th Motorized Division, Fort Leonard Wood, Mo.; **JAMES T. McKNIGHT,** lieutenant (jg), Office of Naval Inspector of Ordnance, Navy Department, Providence.

DENNIS BROWN PEYTON, formerly at the Naval Aircraft Factory, Philadelphia, is now a lieutenant in the U. S. Navy.

BRIG.-GEN. BURTON O. LEWIS has been transferred from duty with the Office of the Chief of Ordnance in Washington, to the rank of Chief, Boston Ordnance District.

Organize Peoria SAE Group

SAE members in Peoria have formed the Peoria Group of SAE, and elected G. E. Burks, chief engineer of Caterpillar Tractor Co. (seated at the left) its first chairman. E. W. Jackson, Jr., chosen vice chairman, is seated with Mr. Burks. Standing, left to right, are C. S. Stone, elected activities chairman, and C. H. Paul, secretary - treasurer of the new Peoria Group

Courtesy of Peoria "Star"

APPLICATIONS Received

The applications for membership received between Oct. 15, 1942, and Nov. 15, 1942, are listed below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

Baltimore Section

Hammond, Lloyd J., managing partner, Hammond & Seidel, Baltimore.

Jarrett, Tracy C., chief metallurgist, American Hammered Piston Ring Division, Koppers Co., Baltimore.

Strobel, Everett H., Lt., U. S. Army, Ordnance Department, Aberdeen Proving Ground, Md.

Theobald, Albert L., owner, Theobald Automotive Co., Baltimore.

Buffalo Section

Bailey, Stanley J., chief draftsman, Bell Aircraft Corp., Buffalo.

Canadian Section

Orr, Frederick James, manager, Walkerville Branch, Somerville, Ltd., Walkerville, Ont., Canada.

Ursaki, George L., Somerville, Ltd., Walkerville, Ont., Canada.

Chicago Section

Balk, Leon J., mechanical engineer, U. S. Army Air Forces, Buick Aircraft Engine Plant, Melrose Park, Ill.

Eilitz, Arthur H., field engineer and service representative, war products, A. C. Spark Plug Co., Flint, Mich. Mail: 3226 Hartzell St., Evanston, Ill.

Haase, Frank W., superintendent of motor equipment, Illinois Bell Telephone Co., Chicago.

Johnston, Raymond W., superintendent of maintenance, Suburban Transit System, Inc., Worth, Ill.

Jones, Harold F., chief lubrication engineer, Cities Service Oil Co., Chicago.

Krupp, Robert F., single cylinder project engineer, Dodge Division, Chrysler Corp., Chicago.

Osborne, Lyle Edwin, experimental engineer, Bendix Products Division, Bendix Aviation Corp., South Bend, Ind.

Cleveland Section

Adler, Malcolm S., project engineer, S. K. Wellman Co., Cleveland.

Berg, Herbert H., tool designer, Weatherhead Co., Cleveland.

Brash, Frederick L., layout draftsman, Cleveland Diesel Engine Division, General Motors Corp., Cleveland.

Courtot, Louis B., mechanical engineer, Weatherhead Co., Cleveland.

Crede, Richard E., tool engineer, Weatherhead Co., Cleveland.

Duffy, Michael J., general foreman, Weatherhead Co., Cleveland.

Edgerton, Alfred Dixon, patent counsel, White Motor Co., Cleveland.

Fenker, Grafton F. J., lubrication technician, Pennsylvania Refining Co., Cleveland.

Finkelmann, John L., chief chemist, The Warren Refining & Chemical Co., Cleveland.

Gearheart, Don H., vice-president and general manager, The Warren Refining & Chemical Co., Cleveland.

Gibbons, Chester R., development engineer, War Department, Cleveland Ordnance District, Cleveland.

Gruss, Thomas J., tool designer, Weatherhead Co., Cleveland.

Judd, Charles Henry, chief engineer, Tinnerman Products, Inc., Cleveland.

Kristoff, Louis, chief chemist, National Aluminum Cylinder Head Co., Cleveland.

Kuhen, Floyd George, Jr., electrical furnace engineer, S. K. Wellman Co., Cleveland.

Lester, Thomas J., plastic and tool engineer, Lester Engineering Co., Cleveland.

Liss, Irwin, plant engineer, Thompson Products, Inc., Euclid, Ohio.

Lowey, Francis John, project engineer, S. K. Wellman Co., Cleveland.

Mack, Thomas J., tool designer, Weatherhead Co., Cleveland.

Mehollin, Fred W., cost estimator, Weatherhead Co., Cleveland.

Mercer, A. L., assistant to the president, Cleveland Tractor Co., Cleveland.

Mlinar, Elmer J., assistant superintendent, Weatherhead Co., Cleveland.

Newgren, Elmer A., chief of time study, Weatherhead Co., Cleveland.

Otto, Howard, works manager, A. W. Hecker, Cleveland.

Phillips, Edward A., metallurgist, National Bronze & Aluminum Products Co., Cleveland.

Schoepfle, Karl C., methods supervisor, Weatherhead Co., Cleveland.

Shier, Gary A., general foreman, Weatherhead Co., Cleveland.

Smith, Oscar W., automotive engineer, Socony-Vacuum Oil Co., Inc., Detroit. Mail: 16712 Westdale Ave., Cleveland.

Stier, Frank C., factory manager, Baker-Raulang Co., Cleveland.

White, Frank Fretter, assistant to vice-president, Weatherhead Co., Cleveland.

White, Nelson E., manager, lubricating sales, The Warren Refining & Chemical Co., Cleveland.

Dayton Section

Ahlers, Joseph A., editor, Automobile Digest Publishing Co., Cincinnati, Ohio.

Prosser, Thomas W., lubricating engineer, Sinclair Refining Co., Columbus, Ohio.

Rosner, Aaron Harold, assistant aeronautical engineer, U. S. Army Air Forces, Wright Field, Dayton.

Detroit Section

Agren, Douglas E., engineer, Industrial Wire Cloth Products Corp., Wayne, Mich.

Allbright, Rex, project engineer, Detroit Diesel Engine Division, General Motors Corp., Detroit.

Anderson, Chester B., junior engineer, Chevrolet Gear & Axle Division, General Motors Corp., Detroit.

Blaisdell, Charles C., lubrication engineer, Penola, Inc., Detroit.

Blenkush, Philip George, instructor, University of Detroit, Detroit.

Brams, Stanley Howard, Detroit editor, The Iron Age, Detroit.

Campbell, Frank J., field engineer, Cadillac Motor Car Division, General Motors Corp., Detroit.

Crary, John D., service engineer, Freeman Chemical Co., Detroit.

Domal, Stanley Alan, chief draftsman, Long Mfg. Division, Borg-Warner Corp., Detroit.

Hanley, George P., draftsman, Long Mfg. Division, Borg-Warner Corp., Detroit.

Huck, Walter J., engineer, Long Mfg. Division, Borg-Warner Corp., Detroit.

Hudson, Duncan G., design engineer, Continental Aviation & Engineering Corp., Detroit.

Jones, Robert S., production control supervisor, Packard Motor Car Co., Detroit.

Knowles, James, design analysis, Packard Motor Car Co., Detroit.

Lysett, Daniel W., Jr., chief draftsman, Long Mfg. Division, Borg-Warner Corp., Detroit.

Maki, Edwin Carl, laboratory technician, Cadillac Motor Car Division, General Motors Corp., Detroit.

McCuen, Newell H., inspection foreman, Cadillac Motor Car Division, General Motors Corp., Detroit.

Misch, Herbert L., design analysis, Packard Motor Car Co., Detroit.

Partel, Stanley C., Jr., engineering follow-up, Detroit Diesel Engine Division, General Motors Corp., Detroit.

Petrik, John J., vice-president, Machinery Design, Inc., Detroit.

Reigner, Hal M., test engineer, Ford Motor Co., Dearborn, Mich.

Rodger, William Robert, laboratory engineer, Chrysler Corp., Detroit.

Schuett, Bromley B., research engineer, Long Mfg. Division, Borg-Warner Corp., Detroit.

Trese, Ralph H., vice-president, Timken-Detroit Axle Co., Detroit.

Indiana Section

Baker, Gerald W., checker, Marmon-Herrington Co., Inc., Indianapolis.

Duesenberg, Denny D., charge of experiments, Marmon-Herrington Co., Inc., Indianapolis.

McCrady, Harry E., assistant chief engineer, Marmon-Herrington Co., Inc., Indianapolis.

McCuen, Marshall D., production engineer, Allison Division, General Motors Corp., Indianapolis.

Newsom, Nathan H., technical analyst, Cummins Engine Co., Columbus, Ind.

Wahl, Clarence William, chief engineer, Hicks Body Co., Inc., Lebanon, Ind.

Metropolitan Section

Alsobrook, Benjamin Russ, service engineer, Wright Aeronautical Corp., division of Curtiss-Wright Corp., Paterson, N. J.

Anderson, Axel Warner, engineering trainee, Wright Aeronautical Corp., division of Curtiss-Wright Corp., Paterson, N. J.

Baer, Hermann W., engineer and foreman, Engineering Service Co., Long Island City, N. Y.

Carpenter, John R., tool designer, Durst Productions Corp., New York City.

Condon, James Stevens, engineering trainee, Wright Aeronautical Corp., division of Curtiss-Wright Corp., Paterson, N. J.

Dement, Marvin Ernest, engineering trainee, Wright Aeronautical Corp., division of Curtiss-Wright Corp., Paterson, N. J.

DuFlon, Alton H., Jr., The B. G. Corp., New York City.

Ellenson, Robert B., tool designer, Durst Productions Corp., New York City.

Hegge, Joseph I., Jr., student engineer, Wright Aeronautical Corp., division of Curtiss-Wright Corp., Paterson, N. J.

Henderson, Sherwood William, engineering trainee, Wright Aeronautical Corp., division of Curtiss-Wright Corp., Paterson, N. J.

Hodgson, Robert F., senior test engineer, Wright Aeronautical Corp., division of Curtiss-Wright Corp., Paterson, N. J.

Marchev, Alfred, vice-president, Republic Aviation Corp., Farmingdale, L. I., N. Y.

Schutz, Paul, chief engineer, The Heil Co., Hillside, N. J.

Stewart, Alex, director of research, National Lead Co., New York City.

Thornley, William Edward, field engineer, Wright Aeronautical Corp., division of Curtiss-Wright Corp., Paterson, N. J.

Veterito, A. Michael, production engineer, Brewster Aeronautical Corp., Johnsville, Pa. Mail: 361 90th St., Brooklyn, N. Y.

Vignini, Walter Robert, tool coordination, Eclipse Aviation, division of Bendix Aviation Corp., Bendix, N. J.

Wyllie, John W., Jr., superintendent of maintenance, Seaboard Freight Lines, Inc., Bridgeport, Conn. Mail: 35 Perry St., Fairfield, Conn.

Milwaukee Section

Duquemin, Charles Henry, assistant production engineer, Le Roi Co., Milwaukee.

Muskegon Group

Snyder, Taylor Milton, chief metallurgist, Continental Motors Corp., Muskegon.

New England Section

Cochrane, Kingsley, foreman, Colonial Beacon Oil Co., Everett, Mass.

Stone, Clarence H., New England distributor, Briggs Clarifier Co., Washington, D. C. Mail: 38 Brookside Ave., Winchester, Mass.

Northern California Section

Butcher, Robert T., test mechanic, Hall-Scott Motor Car Co., Berkeley, Calif.

Gunderson, Glen R., service engineer, Charles W. Carter Co., San Francisco.

Swan, **Robert, Capt.**, U. S. Army, automotive officer, 101st C. A. Brigade (AA) San Francisco.

Northwest Section

Craig, **Henry C.**, sales engineer, Turco Products, Inc., Los Angeles, Calif. Mail: 1020 Fourth Ave. S., Seattle, Wash.

Sanders, Walter F., automotive advisor, U. S. Army, Ordnance Department, Tacoma, Wash.

Oregon Section

First, **J. L.**, shop foreman, Consolidated Freightways, Inc., Portland.

Peoria Group

Eger, **George William, Jr.**, research engineer, Caterpillar Tractor Co., Peoria.

Fleck, Kenneth J., research engineer, Caterpillar Tractor Co., Peoria.

Grim, George B., research laboratory engineer, Caterpillar Tractor Co., Peoria.

Millan, John F., research engineer, Caterpillar Tractor Co., Peoria.

Kranc, Stanley John, research engineer, Caterpillar Tractor Co., Peoria.

Robinson, R. R., supervisor of engine design, Caterpillar Tractor Co., Peoria.

Sausser, James R., Jr., research engineer, Caterpillar Tractor Co., Peoria.

Smith, John Hecker, research engineer, Caterpillar Tractor Co., Peoria.

Wallace, William A., junior engineer, Caterpillar Tractor Co., Peoria.

Philadelphia Section

Keller, Edwin H., service engineer, E. I. du Pont de Nemours & Co., Inc., Wilmington, Del.

Pomeroy, Andrew L., automotive engineer, Atlantic Refining Co., Philadelphia.

Scofield, James M., engineer, Jacobs Aircraft Engine Co., Pottstown, Pa.

Swailes, Earl Jack, senior inspector, USN, Bureau of Aeronautics, Brewster Aeronautical Corp., Long Island City, N. Y. Mail: 6955 Ogontz Ave., Philadelphia.

St. Louis Section

Reid, Albert C., owner, Reid Auto Parts, St. Louis.

Southern California Section

Berger, Emric W., chief of manufacturing research, Consolidated Aircraft Corp., San Diego, Calif.

Bixler, Ralph Wesley, engineering checker, Douglas Aircraft Co., Inc., Los Angeles.

Crabtree, James T., junior layout draftsman, Ryan Aeronautical Co., San Diego, Calif.

Dick, Wallace L., layout draftsman, Kinner Motors, Inc., Glendale, Calif.

Hammond, D. C., president, Pacific Fabricating Co., Huntington Park, Calif.

Jernberg, Albert William, tool designer and engineer, Vega Aircraft Corp., Burbank, Calif.

Johnson, William Harold, U. S. Spring & Bumper Co., Los Angeles.

Lawrence, James W., Jr., junior mechanical engineer, National Supply Co., Torrance, Calif.

Lee, Rowland C., assistant chief engineer, Harvill Corp., Los Angeles.

Livermont, Frank W., vice-president in charge of engineering, Tubing Seal-Cap, Inc., Los Angeles.

Lockett, W. J., service engineer, Douglas Aircraft Co., Inc., Santa Monica, Calif.

Moore, W. D., hydraulic engineer, Douglas Aircraft Co., Inc., El Segundo, Calif.

Mosher, Cleighton N., drafting, Gilfillan Bros. Inc., Los Angeles.

Nordlinger, Louis S., Jr., junior weight engineer, Lockheed Aircraft Corp., Burbank, Calif.

Racicot, N. A., Major, U. S. Army, Camp Young, Indio, Calif.

Ricklefs, James S., engineering instructor, Aero Industries Technical Institute, Los Angeles.

Roe, Homer Davis, engine test engineer, Hughes Aircraft Co., Culver City, Calif.

Sander, Edward G., service representative, Ryan Aeronautical Co., San Diego, Calif.

Scott, Hugh F., superintendent, Frazier Wright Co., Los Angeles.

Stavneak, Edward S., carburetor instructor, Curtiss-Wright Technical Institute, Glendale, Calif.

Southern New England Section

Krasnitsky, John, layout man, Pratt & Whitney Aircraft Co., division of United Aircraft Corp., East Hartford, Conn.

Litchfield, Arthur K., sales engineer, Raybestos Division, Raybestos-Manhattan, Inc., Bridgeport, Conn.

Rittweger, John Daniel, layout man, Vought-Sikorsky Aircraft Division, United Aircraft Corp., Stratford, Conn.

Southwest Group

Klein, Louis, Jr., instructor, vocational education, Tulsa Public Schools, Tulsa.

Turnquist, Edwin W., chief engineer, Braden Winch Co., Tulsa.

Syracuse Section

Hecht, Marco, president and general manager, General Products Corp., Union Springs, N. Y.

Rice, Reginald I., Jr., assistant project engineer, Aircooled Motors Corp., Syracuse.

Seavey, Gordon Crowell, special engineer, Aircooled Motors Corp., Syracuse.

Texas Section

Besserer, Carl William, Jr., instructor, mechanical engineering, The University of Texas, Austin, Texas.

Graves, John L., design draftsman, Guiberson Diesel Engine Co., Dallas.

Jackson, Edwin H., secretary, Globe Aircraft Corp., Fort Worth, Texas.

Shapiro, Eli, project engineer, Guiberson Diesel Engine Co., Dallas.

Smith, D. E., Jr., draftsman, Guiberson Diesel Engine Co., Dallas.

Thaheld, Fred E., research engineer, Guiberson Diesel Engine Co., Dallas.

West, John Stark, draftsman, Guiberson Diesel Engine Co., Dallas.

Washington Section

Lombard, Emmons S., captain ammunition inspection, War Department, Office, Chief of Ordnance, Washington, D. C.

Outside of Section Territory

Cochrane, George Stuart White, salesman, Apex Oil Products Co., Minneapolis, Minn.

Cragg, Arthur C., Jr., sales engineer, Donaldson Co., Inc., St. Paul, Minn.

Cresswell, Roger R., planning engineer, Donaldson Co., Inc., St. Paul, Minn.

Donaldson, Frank Arthur, Jr., project engineer, Donaldson Co., Inc., St. Paul, Minn.

Ellinwood, Raymond S., vice-president and general manager, Huntington Precision Products Division, Adel Precision Products Corp., Huntington, W. Va.

Farmer, Herbert F., Lt.-Col., assistant commandant, U. S. Army, Ordnance Motor Transport School, Atlanta, Ga.

Hagglund, Herbert N., assistant general manager, Motor Supply, Ltd., Honolulu, Hawaii.

Howard, John R., project engineer, landplane trainers, Pan American Airways System, Miami, Fla.

Huff, Lee, Jr., Lt., U. S. Army, Ordnance Department, Fort Crook, Nebr.

Kelley, Ellery E., Major, U. S. Army, Fort Crook, Nebr.

Krause, Robert James, service representative, Vultee Aircraft, Inc., Nashville, Tenn.

Manson, Lidia (Miss) research assistant, engineering experiment station, Pennsylvania State College, State College, Pa.

Neyhart, Amos Earl, road training consultant, American Automobile Association Administrative Head, Institute of Public Safety, Pennsylvania State College, State College, Pa.

Thornblad, Wilford T., associate mechanical engineer, U. S. Public Roads Administration, San Francisco, Calif. Mail: Fort St. John, B. C., Canada.

Vallee, Earle Creighton, chief draftsman, Clark Ruse Aircraft Ltd., Dartmouth, N. S.

Winchell, Guilbert S., Lt., USNR, Box 1358, Naval Air Station, Pearl Harbor, Hawaii.

Wise, Clarence E., master mechanic, E. B. Badger & Sons Co., Point Pleasant, W. Va.

Foreign

Robson, John Henry, service engineering director, British Overseas Airways Corp., Bristol, England.

NEW MEMBERS Qualified

These applicants who have qualified for admission to the Society have been welcomed into membership between Oct. 15, 1942, and Nov. 15, 1942.
The various grades of membership are indicated by: (M) Member; (A) Associate Member; (J) Junior; (Aff.) Affiliate Member; (SM) Service Member; (FM) Foreign Member.

Canadian Section

Godwin, Gordon (M) logging engineer, Quebec North Shore Paper Co., 680 Sherbrooke St., W., Montreal, Que.

Cleveland Section

Koffel, William K. (J) junior engineer, National Advisory Committee for Aeronautics, Cleveland (mail) 16912 Bradgate.

Dayton Section

Sawdon, Will W. (J) assistant project engineer, Wright Aeronautical Corp., Division of Curtiss-Wright Corp., Lockland, Ohio (mail) 6714 Sampson Lane, Silverton, Ohio.

Detroit Section

Cooperrider, Harold M., Capt. (SM) assistant chief, tank branch, U. S. Army, Detroit Ordnance District, 1832 National Bank Bldg., Detroit (mail) 16510 Muirland.

Costello, John R. (J) associate automotive engineer, U. S. Army, Ordnance Department, Tank and Automotive Center, Fisher Bldg., Detroit (mail) 11735 Broadstreet.

Nair, Frank B. (M) chief engineer, Falls Spring and Wire Co., Detroit (mail) 8635 Conant Rd.

Shatagin, John T. (M) assistant chief engineer, Goodyear Aircraft Corp., Akron, Ohio (mail) 467 Chalmers, Detroit.

Metropolitan Section

Anderson, William G. (J) instructor of mechanical engineering, Newark College of Engineering, 367 High St., Newark, N. J. (mail) 91 Third Ave.

Fuller, Frederic Edwin (M) chief stress analyst, Ranger Aircraft Engines, Division of Fairchild Engine & Airplane Corp., Farmingdale, L. I., N. Y.

George, Charles M. (J) junior experimental test engineer, Wright Aeronautical Corp., Division of Curtiss-Wright Corp., Paterson, N. J. (mail) 44 Mt. Washington Dr., Clifton, N. J.

Golzio, Ralph John (M) assistant project engineer, Wright Aeronautical Corp., Division of Curtiss-Wright Corp., Paterson, N. J. (mail) 8-05 Sixth St., Fairlawn, N. J.

Junge, Harold P. (A) supervisor of transportation, General Baking Co., 420 Lexington Ave., New York City (mail) 104-72 129th St., Richmond Hill, L. I., N. Y.

Lamb, Daniel Harrison (J) junior test engineer, Wright Aeronautical Corp., Division of Curtiss-Wright Corp., Paterson, N. J. (mail) 16 Lee Ave., Glen Rock, N. J.

Rowe, William Henry, Ensign (J) U. S. Navy, Office, Supervisor of Shipbuilding, 11 Broadway, New York City.

Sherwin, Frederick S. (J) test engineer, Wright Aeronautical Corp., Division of Curtiss-Wright Corp., Paterson, N. J. (mail) Apt. G-32, Abbott Court, Fairlawn, N. J.

Stuart, Carl Wesley (J) junior test engineer, Wright Aeronautical Corp., Division of Curtiss-Wright Corp., Paterson, N. J. (mail) 340-B Plaza Rd., Radburn, N. J.

Zecca, Robert J. (J) junior test engineer, Wright Aeronautical Corp., Division of Curtiss-Wright Corp., Paterson, N. J. (mail) 242 DeKalb Ave., Brooklyn, N. Y.

Northern California Section

Coe, Francis (J) junior engineer, Joshua Hendy Iron Works, Sunnyvale, Calif. (mail) 620 Creek Dr., Menlo Park, Calif.

Dunbar, Gerald C., Major (SM) U. S. Army, Quartermaster Corps, Stockton Quartermaster Motor Base, Stockton, Calif.

Southern California Section

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Texas Section

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Nixon, John M. (M) vice-president, Continental Motors Corp., Texas Division, Dallas.

Sheridan, Harry A. (A) assistant service manager, Guiberson Diesel Engine Co., P. O. Box 1106, Dallas (mail) 7431 Forest Hills Blvd.

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Outside of Section Territory

Beck, Edwin Charles (M) service engineer, Sealed Power Corp., Muskegon, Mich.

Gates Rubber Co. (Aff.) Denver, Colo. Reps: Corr, Albert C., Detroit; Custer, Kenneth G., manager, belt development division; Mullen, Carl P., development engineer.

Foreign

Cam Gears, Ltd. (Aff.) Biscot Rd. Works, Luton, Beds., England. Reps: Leese, Harold: Pentony, Richard.

Thompson, George Anson (A) manager, overhaul plant, Canadian Pacific Air Lines, Ltd., P. O. Box 734, Vancouver, B. C., Canada.

DESIGN ELEMENTS Affecting SAFETY

by J. WILLARD LORD
Safety Engineer, The Atlantic Refining Co.

WHAT motor-vehicle operators want with respect to safety in vehicle design is summarized in this paper, submitted as a preliminary report of Subcommittee D-7 of the SAE Transportation & Maintenance Activity's program.

Discussing the characteristics of vehicle design that make them hard to drive, fatiguing to operate, difficult to maintain, and disliked by drivers and mechanics, and the remedies thereof, the author takes up first cabs with poor visibility and with insufficient headroom for driver comfort. Continuing, he takes up factors that cause drivers to fall asleep and over-violent pitching of the cabs, and presents the results of a careful study made to find the optimum location for mirrors. Heating and ventilation, location and operation of controls with emphasis on braking and safe vehicle lighting are other important subjects discussed in this paper.

■ There is no standard human being. A long-waisted driver in a "standard" cab must hunch over the wheel in order to see properly

IN discussing safety there often is an attitude to treat it as a thing apart – as an extra problem to burden engineer and operator. However, a review of industry will show that safety and efficient operation have gone hand in hand, and today, with equipment and trained men at a premium, we can ill afford losses because of needless accidents. As to this paper, it should be made clear that it primarily applies to items involving operating safety as distinguished from structural safety, or the designing of parts to withstand the loads imposed on them. A truck can be a beautiful piece of engineering and stand up under the most severe use but, at the same time its design can be such that it is the very devil to drive, fatiguing to operate, difficult to maintain, disliked by drivers and mechanics, and frequently involved in accident. It is these latter factors which interest us.

About a year ago we presented to our Engineering Department a so-called "Truck Safety Guide," which was a series of notes prepared with the idea that it would help in writing truck specifications, and would aid the designing engineer to incorporate design elements which would lend themselves to safe operation of the units. I shall quote from this guide and also comment on some of the recommendations made.

Cab Visibility – Great attention is being given to see that drivers have good eyesight as an essential to safe operation. Thus design of cabs is extremely important as regards visibility.

1. Avoid low overhanging cab roofs.
2. Avoid wide pillar posts spreading out at top so prominently as to interfere with range of vision of tall drivers or men long from the waist up.
3. Avoid pillar posts which widen out at base to interfere also with range of visibility, particularly that of short drivers.
4. Avoid the very narrow small cabs which place the driver in a position where he is at definite disadvantage in maneuvering his truck.

Some cab roofs are so low in their relation to the driver's seat that the tall driver is obliged to hunch over in order

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 2, 1941.]

to look out and, while the pillar posts on these cabs may have rather small cross-section in the middle of the posts, they flare out at the top and at the bottom and present a definite obstruction to looking right and left – an obstruction so large as easily to obscure a vehicle approaching an intersection.

Windshield Wipers and Defrosters –

1. Cab shall be provided with built-in windshield defroster equipment.

2. Windshield vents shall not issue hot air in summer. Provide means to close these off.

3. Windshield shall be equipped with right and left wipers.

4. Windshield shall be equipped with sun shade so as to give protection to the front and sides.

Driver Comfort and Its Relation to Fatigue – In digging into some of the causes of truck accidents, it became apparent that many operators were establishing runs which placed drivers behind the wheel for exceptionally long periods of time, and that drivers were becoming exhausted and going to sleep while driving. However, if one will talk to drivers operating various styles of units, one will find a very definite attitude that some units are far less fatiguing to operate than others. In other words, hours behind the wheel is only one factor in fatigue. Trucks that are hard to steer, fitted with uncomfortable seats, requiring heavy pedal pressure to operate, and cabs poorly ventilated and with poor visibility, are elements which have quite as much to do with fatigue as the time element behind the wheel. Thus the following notes were prepared:

Cab Seating –

1. Stress comfortable seat cushions and A-1 cushion materials. Our impression is that cushions have very short life. Cover materials quickly wear through and inner construction of the cushion breaks down. Such cushions materially contribute to fatigue.

2. Seat design should stress comfortable, well-supported position of driver with ample adjustment to accommodate tall and short men. There should be plenty of leg room for the long-legged man, and plenty of adjustment to accommodate the short driver so that he can reach the pedals readily and still have a good back rest.

3. *Angles and texture of seat and back cushions should be studied.* A seat can be so flat as to cause a driver to slide forward continually, or it can be at such an angle as to be too high in front and create an extremely disagreeable and tiresome pressure on the under part of the driver's legs, tending to interfere with circulation. Back cushions should have correct relation to seat cushions, and not tend to push the driver over the wheel.

Control of Cab Pitching – In some trucks, particularly cab-over-engine models, the entire cab is pitched violently when operating over other than main highways. This pitching should be controlled adequately, probably by shock absorbers of truck size.

Cab Ventilation –

1. Cab shall be substantially tight against fumes from engine.

2. Cab shall be substantially tight and insulated from engine heat.

3. Cab roof shall be substantially insulated from sun's heat and winter's cold.

4. Sponge rubber alone is not a satisfactory material for use as piping around doors and ventilators. It rapidly deteriorates in the presence of truck washing compounds, and it is only a matter of weeks before it begins to disintegrate.

5. Cab shall be ventilated adequately at driver's feet as well as above.

6. Provision should be made so that cab can be pressure-ventilated in rain and snow, and thus prevent fogging of windshield.

7. Cab shall be provided with heat for winter. There are those who have taken the attitude that heaters make the driver drowsy, but more and more it is being realized that a comfortable driver will do a better job of driving than one half frozen.

Safety Provisions for Getting Into and Out of Cab –

1. Cab and fender steps should preferably be of Morton or similar tread design, or step plates of Morton tread should be attached.

2. Where stirrups are used, precaution should be taken to prevent a man's foot from going through the stirrup and thus hanging him by his foot; also to prevent a man's heel catching in the stirrup. (It is the Company's practice today to attach to the standard cab steps or running boards, light-weight Morton tread in order to provide secure footing.)

3. Ladders to and from walkways shall have minimum width between side rails of 12 in. The ladder shall be covered or otherwise protected to prevent hitch-hikers from riding the truck.

Mirrors –

1. Cabs shall be equipped with bright, right- and left-hand mirrors. Plain left mirrors and convex right mirrors shall be mounted above or below the normal range of vision. The exact point shall be determined from a complete assembly drawing of truck and tank. Often it is possible to mount the mirror low enough so as to be able to see under the tank.

2. Mirror brackets shall be substantially mounted, preferably on the cab rather than on cab doors. Jiggly hinge-pin mountings should not be permitted.

3. Bracket mountings should be of universal design to give range of adjustment for short and tall drivers.

4. Mirrors themselves shall have a ball-and-socket mounting to permit easy adjustment to meet the requirements of individual drivers.

I feel that far too many operators are not paying enough attention to mirror equipment, and that they would help themselves considerably if they would do so. In fact, I recommend that operators equip their own passenger cars with side mirrors to help them understand the problems of drivers who have to depend entirely on side mirrors in order to see what is in back of them. If one will do this, he quickly will appreciate the disadvantage of the left-hand mirror located in the normal range of vision to the left, and will definitely appreciate the advantage of having this mirror either above or below the normal range of vision.

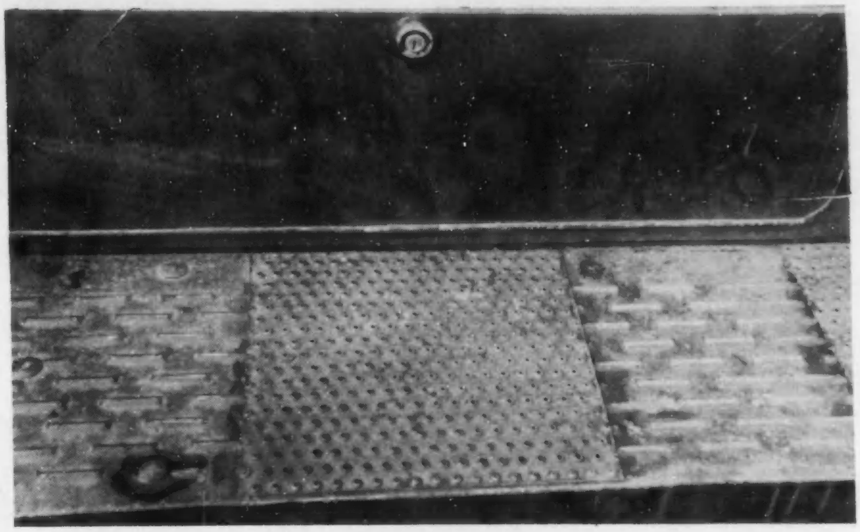
Placing the mirror below the horizontal often facilitates not only looking back, but also looking under the body, making it possible to see a passenger car immediately back of the truck. In fact, I have seen trucks on the road with a rear-vision mirror mounted low on the fender so that the driver could look under a large van body and know if there was anyone immediately behind.

The combination of a plain mirror on the left and a convex mirror on the right works out advantageously. A plain

mirror next to the driver gives him a true picture of what is behind him and how far away it is. The convex mirror on the right presents a wide-angle picture of what is on the right-hand side, and this type of mirror will still tell a good story when mounted on a semi-trailer making a right turn. A plain mirror under these circumstances only shows the right-hand side of the body and becomes useless.

■ Lighting

In view of the fact that operators are trying to get the greatest value out of their investment, it is only natural to expect that they are running their equipment on two or three shifts out of the 24 hr wherever possible. Therefore, it stands to reason that trucks should be equipped with the



■ Typical pressed steel cat running board with Morton tread attached to prevent "slip-and-fall" accidents

best of lighting facilities to help the driver to see where he is going at night and without endangering his equipment or other users of the highway.

The sealed-beam headlight has introduced considerably more light on the highways at night and, in order that truck drivers can see at all in the face of the oncoming glare of these high-candlepower lamps, their trucks should be equipped with similar lights.

The relatively lower speeds of trucks is no reason for assuming that lower-candlepower lamps should be satisfactory. Those who have made most complete studies of night blindness point out and can demonstrate that, in order to be able to see at all in the face of intense glare, well-directed, intense illumination is required. Not only do sealed beams give the desired intensity of light, but they surely overcome the difficulties of poor lighting due to slow accumulation of dust, tarnishing reflectors, and half-burned-out bulbs.

Headlights – Headlights should be of sealed-beam design for present conditions met on the highways and to maintain a standard of excellence of headlamp illumination.

Stop Lights – These lights should be equipped with amber lenses in preference to red, and have 21-cp lamps. They should be of 5- or 6-in. diameter instead of 3-in. so often used. These larger lights are becoming quite common, and are often seen on buses.

Fog Lights –

1. Trucks operating in areas where obliged to drive under foggy conditions should be equipped with fog lights. These lights should be of clear-lens type and mounted low – 20 in. above ground.

2. Fog lights should be connected through a separate fuse so that, if the main light fuse blows, the fog lights will not be affected.

Clearance Lights – Front clearance lights should be lo-

Truck Controls –

1. It is very desirable to have standard arrangement of controls. Many drivers, relief drivers particularly, are called on to drive several makes and styles of trucks.

2. Elaborate and non-standard gearshift arrangements should be explained by lithographed plate on dash. "Decals" quickly disappear.

3. There should be a standard arrangement of pedals and accelerator in the order named: clutch, brake, and accelerator. We do not recommend arrangements as found on some trucks of placing the accelerator between clutch and brake pedals and very close to driver. This position is uncomfortable and cramps the driver's leg.

With regard to an uncomfortable position of the accelerator, drivers will not refuse to drive these trucks; on the other hand, examination of floorboards shows a regular groove worn in the boards by the edge of the driver's shoe, clearly indicating that a poorly placed accelerator pedal forces the driver into an uncomfortable and cramping position. Furthermore, we have had some direct head-on contacts with stationary objects within our own properties where the driver's only explanation was that his foot was cramped and he got his feet twisted.

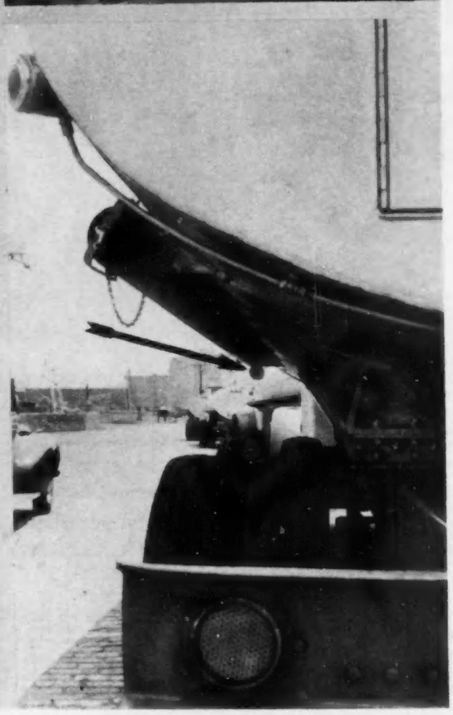
Pedal Pads – Rubber pedal pads are not recommended when any oily condition exists. We find many trucks where drivers have tied cloths over the rubber pads in order to prevent slipping. We recommend calling for Neoprene pedal pads or metal pedal pads with a small lip on the outer edge to prevent the foot from slipping off.

Hand-Brake Lever – We recommend a standard hand-brake lever, easily reached. It should be mounted in a standard position and operated by pulling back. On some trucks the hand brake requires pulling up on the lever to set the brake. This operation is awkward and difficult, and surely lends itself to having a runaway truck.

■ High adjustment of mirror limits vision to the rear

■ Low adjustment of mirror permits seeing under tank and far to the rear

■ Typical mirror adjustment that often is all right for rear vision, but presents a definite blind spot when looking to the left at intersections



■ Same truck as illustrated directly above showing how mirror can reflect a car or pedestrian immediately back of truck

cated so as not to shine into the rear vision mirror and annoy or mislead the driver. It is preferable to mount the front clearance lights high rather than low. A low position of the mirror is desirable as it permits seeing back and under the truck tank to spot a car directly behind.

Directional Switch – Mount this switch under the steering wheel for left-hand operation. The switch should move from left to right so as to aid driver normally in knowing which light he is lighting. All of us undoubtedly have come up behind a truck, seen directional signals pointed one way, and have had the truck turn in the opposite direction. When the directional signal switch is operated forward and backward, this mistake can occur easily.

Low Beam Switch – Mount this switch on toeboard at extreme left. This is more or less a standard mounting today. On older trucks we find the switch still on the dash, and it is practically impossible to find it and still keep one's eyes on the road at night.

Truck Wiring – There have been many dash fires due to defective wiring. They all point out that we need an A-1 wiring job with wire of ample size to carry the lamp load, well insulated and fitted with substantial terminals. Dash instrument connections should be of substantial construction, and fuse blocks should have the best of contacts. Ground returns should be given more attention. The greatest voltage drops all too often are due to poor ground connections in the headlamps and at their mountings.

Front and Rear Bumpers – Recommend consideration to mount auxiliary car bumpers at rear. The front bumper should protect the front fender step.

In connection with this matter of auxiliary car bumpers, particularly at the back of the truck, I believe that, in many cases, it would be feasible to mount a substantial bumper to the bottom of the side members which would serve two purposes: First, it would control backing into loading platforms, and so on, and secondly, it would be low enough to contact car bumpers and thus overcome a perpetual headache of paying for radiator grilles and sets of headlamps. Backing accidents run from 10% to 20% of all accidents in many fleets.

Package Carriers – Tools, Chains, Fittings, Rags –

1. Packages shall not be carried in cabs.

2. Adequate provision should be made for carrying rags, fittings, tools and chains. Where no provision is made,

Inspection of trucks will reveal cabs full of loose rags, chains, and so on.

Horn – Trucks require an A-1 horn and no tin tooter.

■ Fire Protection

Fire-Extinguisher Equipment – Pressure-type extinguishers are recommended. If of the carbon-tetrachloride type, they should have a minimum capacity of ½ gal. If of the dry-powder type (Dugas), the 15-lb unit is recommended.

Truck fires are something to be avoided for, not only are they expensive and often involve loss of life, but in addition they make very unfavorable publicity and are a black eye to the trucking industry. National figures seem to exist only in sketchy form, but here and there some group has made a cross-sectional study and a few figures have been made available.

Fatalities in connection with fire accidents as compared with non-fire accidents are in a ratio somewhat more than three to one. About one out of every five fire accidents involves total loss of both vehicle and cargo. In connection with this study, only about 2½% of the total number of accidents considered were fire accidents, but these few accounted for 23% of the total property damage and accounted for about 8% of the total fatalities. Nearly half of the fires were results of collision of some sort, and of the instances in which it was known whether or not there was gasoline spillage, spillage occurred in nearly three-quarters of the cases. The rather strong inference from this fact is that the gasoline tank was responsible for the spillage in a considerable number of cases, and that had the spillage of tanks been less, the damage and loss of life would have been considerably less. In connection with truck fires, I know of no evidence to show that the ignition, whether off or on, has been a source of ignition for the unwanted fire.

This matter of gasoline spillage and subsequent fires deserves your serious consideration. There used to be a time when discarded hot water boilers were carefully cut in two and used as window boxes for flowers, but now they are made to serve as truck gasoline tanks. Then we see many trucks with light gage tanks mounted on the outside of the frame and without the slightest protection in the event of right angle intersection collision.

Recently the ICC through cooperative effort established structural standards for truck cargo tanks carrying flammable liquids. It is fair to assume that such legislation will go further as accident experience indicates its need and where no action is taken by operators or manufacturers. We should design to minimize the possibility of spillage in the event of collision or other accident, and not wait for legislation to force us to do so.

Aside from fires resulting primarily from collision, there seem to be two other classes – those starting from short circuits in wiring systems, and those associated with leaks in a cooling system using alcohol as an anti-freeze.

Cab fires seem to be associated mostly with some electrical short circuit and often result in burning out the cab. Many of the materials for finishing and dressing up cab appearance seem quite combustible and furnish excellent fuel for fires of this kind.

■ Springs

Springs wear out and break, and usually are operated until they fail. Under normal circumstances, with the vehicle in motion, breakage of a front spring should not interfere with steering and take the truck off the road. However, this kind of accident has been definitely associated with some model trucks, and never with others. The accidents point out an element of design that deserves careful study in connection with developing any new model.

■ Brakes

I have left this topic of brakes until last because it is a big topic, and can be looked at from many angles.

Truck Brakes –

1. Brakes should be of reaction type for smooth application, and brakes should be balanced.

2. Where tractors are purchased from one manufacturer and trailers from another, the assembled unit should have all brakes balanced before delivery to the operating department.

■ A big truck with a very small (3 in.) combination tail and stop light. Such small stop lights are not likely to be observed in daylight

3. Hand brakes should be of standard design adequate to hold the truck and not release when using power take-off. Chock blocks should not be necessary.

4. The hand-brake hook-up should not rely on air brakes set by the hand brake.

Brake control should lend itself to easy, smooth application and balanced distribution of braking effort. This smooth application has been largely accomplished on passenger cars – though on some models there is too much servo action to please me – but on many trucks no such smoothness is found, and the highways bear much testimony to this, as is seen by the repeated marks of sliding tires of one or more wheels. Brakes that perform in this manner are conducive to skidding and side-swipe accidents.

Brakes are receiving considerable attention on the part of regulatory bodies: specifically, the ICC and now the Public Roads Administration, and I am glad they are, for I feel about brakes very much as the old colored driver

with his model T who remarked: "If yo' can't start, there yo' is, but if yo' can't stop, where is yo'?"

I look upon the ICC report released in March, 1941, as very much a preliminary report, forcefully calling to the attention of all that there are too many trucks operating on the highways with brakes incapable of controlling the truck, failing to meet legal requirements, and that something should be done about it.

I also feel that the present study being made by the Public Roads Administration is going to back up the ICC report, and will call for two things:

1. Better design of brakes for better operation and to facilitate maintenance.

2. Better maintenance of brakes by operators.

On this matter of balancing brakes, "Steve" Johnson of Bendix Westinghouse, presented an excellent paper[1] before the American Transit Association last September. Anyone interested in motor vehicle brakes should read this paper if he has not already done so.

Mr. Johnson points out that small differences of 5 psi in the pressure required to bring the shoes into the drums will account for tremendous differences in the amount of work done by the various brakes in a unit – that the balancing program calls for: first, the thorough cleaning and lubricating of all parts; secondly, adjustment of springs in brake chambers so that they all operate or respond to a uniform pull; and finally, the adjustment of pull-back springs on the various axles so that these are uniform.

To accomplish this job today requires quite a selection of springs, and the balancing operation is very likely to be thrown completely out if a tractor with balanced brakes is put under some other semi-trailer. This would seem to point out a design job – design first to include better lubrication of the working parts of brakes so that we will get away from frozen or partly frozen brake mechanisms, and second, some standardization work so that brake release springs and related parts will have much narrower limits for the pull required to operate them.

There are many other brake items involving design, and I mention one or two for they are indicative that there is still plenty of work for the manufacturer to do in improving design – that brake troubles are not all due to neglect and poor maintenance. Just recently a group of new trucks were found to have short muffler tail pipes and the hot exhaust impinged directly on one of the brake hoses and brake diaphragms, destroying them. Other new trucks of large size could not be held on a slight grade with the hand brake . . . "S" cams were found very soft on some new trailers. The cams were flattened where contact took place with the brake shoe rollers. This flattening made adjustment difficult and brake operation erratic. . . . The rear-axle housings on a group of trucks were not provided with seams to keep the gear oil from running out into the brake drums. The manufacturer was unable to provide a seal due to the construction of the housing, and left the problem entirely up to the operator. The absence of the seal did not present a problem on trucks operated on level streets but, if a truck was parked on a slant or operated on a high crowned road, the oil ran into the brake drum on the low side.

From a maintenance viewpoint, brakes have not been given the attention by operators that they deserve. The brake job is a dirty one, and all too often is assigned to a second-rate mechanic. Practically no one has attempted to teach him about brake maintenance, and he picks up information as best he can – information incomplete in essential details and which is often only partly right.

First we must put an A-1 mechanic on the brake job, give him the necessary brake repair tool equipment, and then teach him real brake maintenance.

But this will do little good unless operating supervisors allow sufficient time to do a first-class job. For instance, many operators pay close attention to the amount of lining on the shoes and will re-line when necessary but, at the time of re-lining, they will be in such a hurry to get the job out, the only thing done is to replace the worn-out lining and, unless the drum is very badly scored, there will be no check or attempt made to re-surface. It is impossible to tell drum condition without accurate measuring devices, and many mechanics will pass the drum as being satisfactory if the surface is fairly smooth. The visual examination will give no indication of a badly tapered drum or one that is out-of-round beyond the figures of good brake-shoe clearances. Thus, only compromise adjustments are possible. Such maintenance methods must be corrected if full life and efficiency are to be obtained from newly re-lined brakes.

I feel that manufacturers and operators have a very large joint responsibility of training maintenance personnel. Manufacturers are the logical source for detailed, accurate, standard repair instructions. They could well take the lead to develop and supply such information in readily digestible shop terms, and prepare these instructions so that they would be placed in tool rooms where they easily could be referred to by mechanics when necessary. Operators could well set up a definite program providing for the training of their maintenance personnel in accordance with these instructions and thus raise the standard of maintenance.

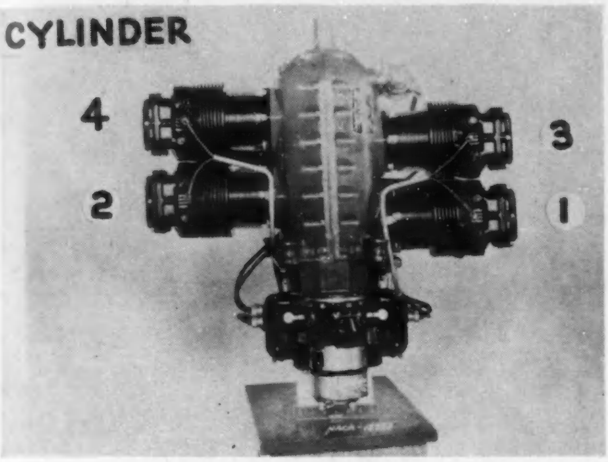
This plan may sound a little visionary, but it seems to me that, in the overall picture, maintenance is definitely a part of the automotive industry, and its badly needed development is primarily the industry's problem. Too many repair instructions practically stop short at telling how a unit works, and fail to define in detail those adjustments and clearances essential to correct re-assembly after the unit has been torn down to replace one or more worn or broken parts.

There is a huge job to be done involving both design and maintenance, and it calls for closer cooperation between manufacturers and operators. It is my sincere hope that such cooperation will flourish under our new Transportation and Maintenance plan of organization developed by our 1941 Vice President and Chairman of the T&M Activity – T. L. Preble.

I have just touched on a few items – not even scratching the surface – but I feel that within the T&M group there should be a far greater exchange of thought and experience, and that this material should accumulate at SAE headquarters and go back to the manufacturers possibly through a general contact set up in the Automobile Manufacturers Association. Some such program should more accurately picture the extent of various difficulties, lead to more prompt revision of design elements which are making trouble, and aid in the development of maintenance instructions which are the logical starting point for any comprehensive plan of training mechanics for better maintenance.

[1] See SAE Transactions, August, 1941, pp. 301-308: "Brakes – Their Analysis and Balancing," by S. Johnson, Jr.

Cooling Characteristics of SUBMERGED LIGHT AIRCRAFT ENGINES



■ Fig. 1 – Top view of Continental A-75 engine

by H. H. ELLERBROCK, JR.
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■ ■ ■

RESULTS are presented of tests on an engine to determine the quantity of air and the pressure difference required for satisfactory cooling at sea level and at altitude. The tests are part of an investigation that has been started by the NACA to determine the performance of a small airplane with two Continental A-75 aircooled engines enclosed in the wings.

The results showed that the engine cooled satisfactorily at sea level with wide-open throttle and maximum power mixture with 4.5 in. of water pressure difference across the baffles and 1.06 lb of cooling air per sec. In addition, from the tests on the ground, calculations showed that the maximum cylinder temperature would not exceed the limit of 500 F at 8000-ft altitude with wide-open throttle and either maximum power or maximum economy mixture if the pressure difference across the baffles was 2.6 in. of water. The cooling air required with this pressure difference would be 3/4 lb per sec. The percentage of the brake horsepower required for cooling with the foregoing conditions at sea level and at 8000-ft altitude would be approximately 1.0 and 1/2, respectively.

■ ■ ■

THE exposed wing nacelles of modern multi-engine airplanes contribute appreciably to the drag of the airplane. The drag of the nacelles, form and cooling drag, is approximately 10 to 25% of the drag of the airplane, depending on the wing thickness. The necessity for reduction of engine-nacelle drag has become increasingly important owing to the gradual elimination of other sources of parasite resistance. An obvious refinement for multi-engine airplanes is the removal of the nacelles from the wings and the enclosure of the complete powerplant within the wing.

An investigation has been started by the NACA to determine the performance of a small airplane with two Continental A-75 aircooled engines enclosed in the wings. One of the main problems with engine installations in the wing is the cooling of the engine. The cylinders must be baffled completely and the cooling-air quantity must be reduced to as low a value as possible. A low quantity of cooling air is necessary in order that the wing ducts may be small so as to cause little interference with the aerodynamic characteristics of the wing.

In normal installations the crankcase and the oil sump of the Continental A-75 engine are cooled by air flowing over them but, when the engine is in the wing, no such cooling is possible. This lack of cooling causes the oil to heat up and introduces another problem: that of providing adequate oil coolers.

The first requirement is the construction of a set of baffles around the cylinders for use in the wing and the determination in the laboratory of the cooling and the performance characteristics of the engine with these baffles. The present paper gives the steps taken in evolving a set of baffles for the Continental A-75 engine suitable for use in the wing of the projected airplane, and the cooling and the performance characteristics of the engine at sea level

[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., March 14, 1941.]

■ Fig. 2 – Top view of baffles

and at altitude with these baffles. In addition, a description of a system used to cool the oil and of tests with this system is given.

The tests were conducted at the Langley Memorial Aeronautical Laboratory of the NACA during the summer of 1940.

■ Description of Equipment

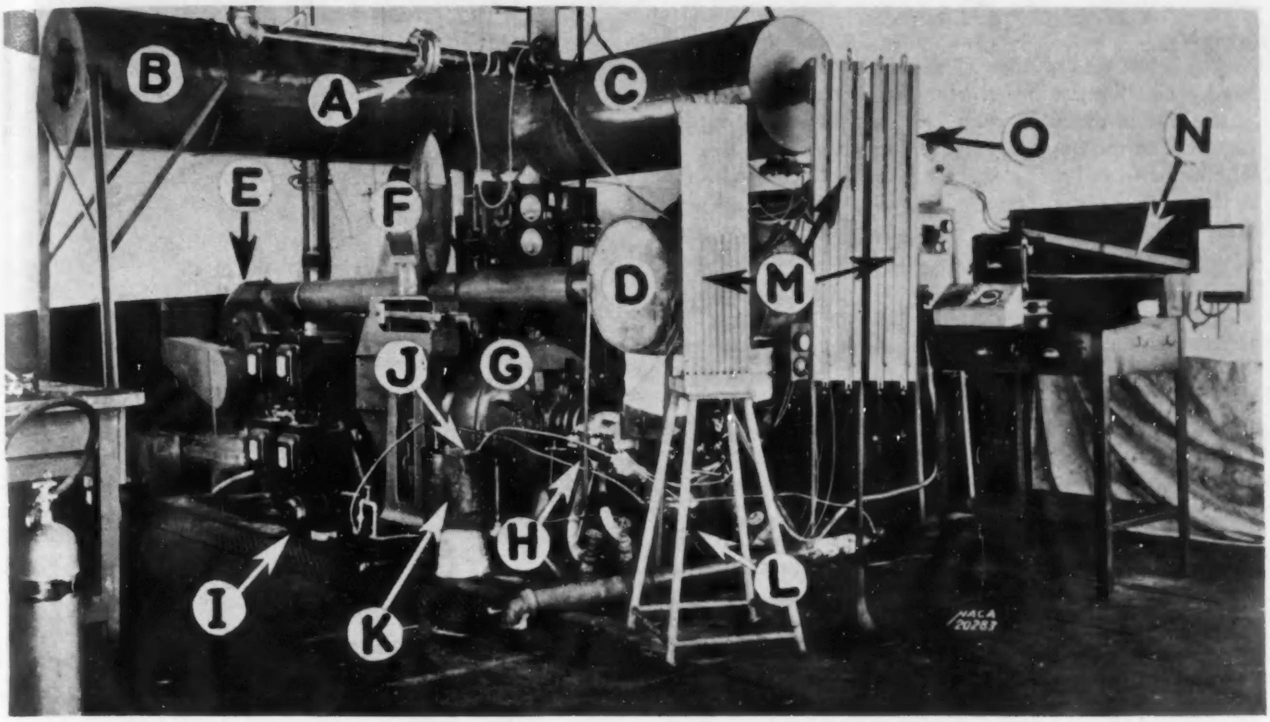
The Continental A-75 engine used in these tests is a 4-cyl, horizontally opposed piston, aircooled engine rated 75 hp at 2600 rpm with 29.92 in. of hg pressure (dry) and 60 F temperature in the intake manifold. The bore and the stroke are 3⅞ and 3⅝ in., respectively, and the compression ratio is 6.3:1. The displacement is 171 cu in. A top view of the engine is shown in Fig. 1. The cylinder numbering shown in the figure is used throughout the paper. The engines in the projected airplane are to be used as pushers and, for that reason, the air is introduced to the cylinders from the end of the engine that is normally the rear. Thus, cylinders 1 and 2 are nearest the leading edge of the wing and cylinders 3 and 4 are nearest the trailing edge of the wing. The engine was mounted in the laboratory on a stand and connected to an electric dynamometer.

The cylinders were enclosed in sheet-aluminum baffles that fitted tight against the fins except at the entrance and the exit of the baffles. It was necessary to make the baffles such that they would not project above or below the crankcase because the wing thickness in the projected airplane is just a little greater than the crankcase thickness. How this feature was accomplished is shown in Fig. 2. The baffles were connected by a smooth passage to a rectangular duct that will be attached in the airplane to a duct extending to the leading edge of the wing. The entrance and the exit areas of a baffle around any one cylinder were approximately 1.6 times the free-flow area between the fins. The baffles, when first made, had many flanged joints held together with bolts. Later, most of the joints were welded, to reduce the leakage, and the welded baffles are the ones shown in Fig. 2. Tests were made with the baffles sealed and open at the rocker boxes and flanges indicated by points *A* and *B*, respectively, in Fig. 2.

The engine set-up is shown in Fig. 3. The cylinder cooling system consisted of an orifice tank, used for measuring the quantity of air, connected to a centrifugal blower, a surge tank, and air ducts between the blower and the surge tank and between the surge tank and the baffles. A surge tank was installed in the combustion-air inlet system above the engine to reduce pulsations, and a thin-plate orifice was used to measure the air consumption. Standard test-engine equipment was used to measure engine speed, fuel consumption, and engine torque.

Iron-constantan thermocouples and a potentiometer were used for measuring the cylinder temperatures. The temperatures of No. 1 cylinder (see Fig. 1 for cylinder locations) were measured with eight thermocouples on the cylinder head, six on the barrel, and one on the flange as shown in Fig. 4. Thermocouples were similarly located

■ Fig. 4 – Location of thermocouples



■ **Fig. 3 – Set-up of apparatus**

A, thin-plate orifice
B, orifice tank
C, inlet-air surge tank
D, surge tank
E, cooling-air blower
F, dynamometer scale
G, dynamometer
H, engine
I, auxiliary oil pump
J, oil cooler
K, water tank
L, oil sump
M, manometers
N, inclined manometer
O, fuel-measuring stand

on Cylinder 4 except that, in place of a thermocouple below the bottom spark plug, as No. 15 thermocouple on Cylinder 1, a gasket-type spark plug was placed under the top spark plug. Also a thermocouple was located in the rocker box of Cylinder 4 (Fig. 4). Thermocouples were placed on Cylinders 2 and 3 at the same locations as thermocouples Nos. 1, 14, and 15 on Cylinder 1.

The temperature of the inlet cooling air was measured in the duct before the baffles. The temperatures of the cooling air at the outlet of each barrel and head were measured separately. Each temperature was measured by a multiple thermocouple consisting of two thermocouples connected in series. Oil-in temperatures and the pressures and the temperatures in the inlet manifold were obtained. Total-head tubes placed in the air ducts before the baffles were used to measure the pressure drop across the cylinders. The pressure in an exhaust pipe 1 in. from the port opening was also measured. This pressure was obtained in order to determine the back pressure on the engine in the laboratory and thus determine whether the engine was delivering rated power.

■ Baffle Tests

The various sections of the first baffles used, as previously mentioned, were bolted together. Tests were conducted on the engine with these baffles to determine the quantity of air required for adequate cooling with wide-open throttle at 2600 rpm. A large gap was left between the baffles and the cylinder wall around the rocker boxes at the point marked *A* in Fig. 2 so that air would flow over the rocker boxes. The baffles at the flanges, point *B* of Fig. 2, fitted tightly against the cylinder wall so that little air flowed over the flanges. Water was sprayed over the tank that formed the oil sump.

The results of these tests with the bolted baffles are given in Table 1.

The highest head, barrel, and flange temperatures are denoted in the table by T_H(max.), T_B(max.), and

Table 1 – Results of Baffle Tests

	Bolted baffles	Welded baffles: Rocker box leaks, Not sealed	Welded baffles: Rocker box leaks, Sealed	Welded baffles: Oil cooler, Without	Welded baffles: Oil cooler, With	Welded baffles: Opened at flanges
I HP	78.5	75.4	79.4	77.5	76.9	75.1
ENGINE RPM	2600	2570	2610	2618	2630	2598
$\Delta P\rho/\rho_0$, IN. WATER	7.3	6.8	7.5	5.7	5.8	7.5
COOLING AIR LB/SEC	1.58	1.38	1.34	1.14	1.15	1.38
OIL-IN, °F	212	202	217	240	200	163
T_H MAX, °F	493	493	496	521	519	483
T_B MAX, °F	362	354	346	368	362	334
T_F MAX, °F	350	348	343	360	351	317
COOLING AIR, °F	94	110	94	100	98	96

T_F(max.), respectively. The highest head temperature with bolted baffles, 493 F, was in the "tunnel" between the rocker boxes of Cylinder 4, that is, in the center of the head. The highest barrel and flange temperatures were at the rear of the same cylinders (at positions corresponding to Thermocouples 4 and 1, respectively, in Fig. 4). The highest head and barrel temperatures were considered satisfactory but the flange temperature was considered a little high, 350 F, and the quantity of air was too great, 1.58 lb per sec. The air was found to be leaking through the bolted joints.

Almost all of the joints in the baffles were then welded and the gaps between the cylinder walls and the baffles at the rocker boxes were reduced to mere slits. The results with these baffles are shown in Table 1. The quantity of cooling air decreased appreciably (from 1.58 to 1.38 lb per sec) compared with what was needed with the bolted baffles. The temperature in the rocker box of Cylinder 4 increased from 276 F with the bolted baffles to 312 F with the welded baffles. This increase was due to the fact that the gap between the baffles and cylinder walls around the rocker boxes had been greatly reduced. All temperatures were considered satisfactory except the hottest flange temperature, which was still high (348 F).

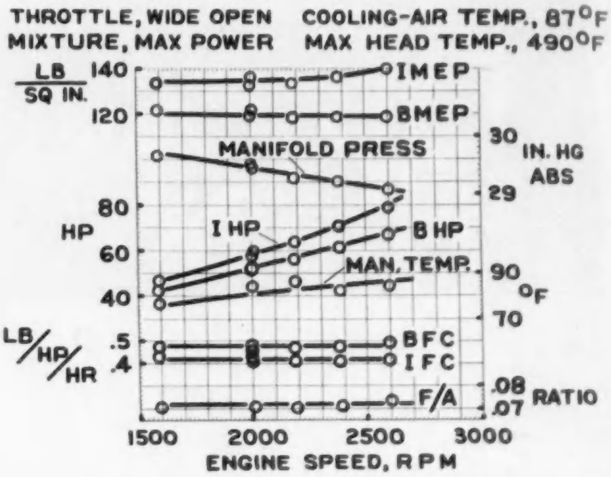
The slits between the baffles and the cylinder walls around the rocker boxes were next closed with a cementing mixture to determine the decrease in cooling-air weight. There was little change in temperatures and the air weight as compared with the previous baffles except that the rocker-box temperature increased from 312 F to 330 F, which increased the oil-in temperature. Neither the oil-in temperature (217 F) nor the highest flange temperature (343 F) was considered satisfactory.

The water spray was next removed from the oil sump and an oil cooler and pump were placed in series with the sump. Oil was circulated from the sump, through the cooler, which was placed in a tank of water as shown in Fig. 3, and back to the sump again. The cooler had a large surface and was installed for the purpose of lowering the oil temperature and, in turn, perhaps the flange temperature. The results of the tests with and without the oil cooler at a lower pressure drop than was used in the other tests are given in Table 1. The oil-in temperature was decreased 40 F and the highest flange temperature was decreased 9 F. There were indications that the flange temperatures would decrease appreciably with further decrease of oil-in temperature.

A test was accordingly made with an oil-in temperature of 163 F and, in addition, the baffles were opened wide at the flanges. The highest flange temperature was 317 F with a pressure drop of 7.5 in. of water. A survey of top and bottom flange temperatures showed that all of the flanges were approximately 250 F except the bottom flanges of Cylinders 2 and 4, which were a little over 300 F. The airweight had been reduced from 1.58 lb per sec with the bolted baffles to 1.38 lb per sec, T_F(max.) from 350 F to 317 F, T_H(max.) from 493 F to 483 F, and T_B(max.) from 362 F to 334 F. This final baffle and oil-cooler set-up with an oil-in temperature of approximately 165 F was used in all subsequent tests.

■ Performance Tests

After the quantity of cooling air had been decreased as much as possible, tests were made to obtain data from which the performance and the cooling characteristics of the engine for any atmospheric condition could be predicted. Tests were conducted at wide-open throttle, maximum power mixture, and with the smallest quantity of cooling air possible with adequate cooling over the range of engine speeds from 1600 to 2600 rpm to determine the variation of mean effective pressures and specific fuel consumptions with engine speed. The results of the tests are shown in Fig. 5. All values of mean effective pressures and horsepowers given in this paper are observed values except those in predicted performance charts. The friction horsepower was determined by motoring the engine at the

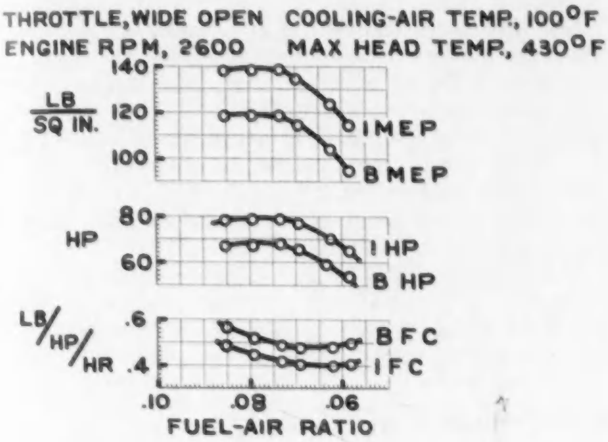


■ Fig. 5 – Variation of performance with engine speed

inlet pressures and speeds used in the power runs. Fig. 5 shows that the manifold pressure was less than atmospheric due to the drop through the air measuring system and that the pressure decreased as the engine speed increased. The "manifold pressure" referred to in this paper was the pressure ahead of the carburetor. For a given manifold pressure and temperature, from the results of Fig. 5, the bmep and imep will increase with increase of speed. The tests were made with a maximum power mixture and the indicated and brake specific fuel consumptions and the fuel-air ratio remained constant over the range of speeds tested.

A series of tests was made at wide-open throttle and 2600 rpm to determine the variation of power and specific fuel consumption for a range of fuel-air ratios; the results of these tests are shown in Fig. 6. The power is a little greater in Fig. 6 at the maximum power mixture than that given in Fig. 5 at 2600 rpm owing to the fact that the engine was a little cooler in the fuel-air ratio tests than in the speed tests. The power remained fairly constant for fuel-air ratios from 0.07 to 0.085 but, below 0.07, it decreased rapidly. The minimum specific fuel consumption was obtained with a fuel-air ratio of 0.065, but a fuel-air ratio of 0.069 will be used hereinafter to denote the maximum economy mixture because the power is appreciably greater with 0.069 than with 0.065 fuel-air ratio. The difference in specific fuel consumption with the two ratios is negligible.

For a given throttle setting, the power that an engine develops when hot usually is increased as the cylinder temperatures decrease. The power required for cooling will increase and it is of interest to determine whether the net brake power (brake power minus blower power for cooling) will increase as the temperatures decrease. The blower power was based on 100% efficiency. Overall efficiencies of wings with ducts as high as 100% have been obtained in wind-tunnel tests with cold air. This phenomenon is caused by the fact that flow through such ducts sometimes affects the external flow about the wing in a beneficial manner. The efficiency is the ratio of the power required to force the air across the engine to the power required by the change in drag of the wing with the wing duct. If hot air is used, the "Meredith effect" will increase the overall efficiency to something greater than 100%. Tests were therefore made at full throttle and 2600 rpm with maximum-power mixture and maximum-economy mixture to find the variation of net brake power with quantity of cooling air. The results showed that, as the pressure drop across the baffles increased from 5.5 to 13.5 in. of water, the net power increased a little and then decreased slightly for the maximum-power mixture. The net power decreased a little for the maximum-economy mixture. The cylinder temperatures decreased appreciably and the conclusion can be made that it is better to operate the engine with a large pressure drop, when the efficiency of the cooling system is 100% or greater, because the net power is about the same as with a low pressure drop and the engine is much cooler, which is advantageous for long life.



■ Fig. 6 – Variation of performance with fuel-air ratio

■ Cooling Tests

Equations have been derived by the NACA[1] in which average head and barrel temperatures of the cylinder are given as functions of the engine and the cooling variables.

The equation for the head may be written as follows:

$$\frac{T_H - T_A}{T_G - T_H} = K_1 \frac{I^n}{(\Delta p\rho/\rho_o)^m} \quad (1)$$

where

T_H, the average cylinder-head temperature, F

T_A, inlet temperature of cooling air, F

T_G, effective gas temperature, F

I, indicated horsepower

Δp, pressure difference across the cylinder, in. water (includes loss out exit of baffle)

ρ, average density of cooling air entering and leaving fins, lb-ft^{-4} sec^2

ρ_o standard density (taken as corresponding to 29.92 in. hg and 70 F in this paper), lb-ft^{-4} sec^2

n and m, exponents

K_1, constant

A similar equation may be written for the barrel. Equations have also been derived for temperatures at individual points on the cylinder head and the barrel. Thus, for a temperature at a given location on the head, the following equation may be written:

$$\frac{(\Delta p\rho/\rho_o)^m}{I^n} \frac{(T_{H_X} - T_A)}{(T_G - T_{H_X})} = F_1\,(\Delta p\rho/\rho_o) \quad (2)$$

where T_{H_X} is the temperature of the cylinder head at some one point. The other symbols have the same significance as in Equation (1), and m, n, and T_G have the same values as in Equation (1). A similar equation may be written for individual barrel temperatures. For most thermocouple locations it has been found that $F_1(\Delta p\rho/\rho_o)$ can be replaced by a constant. The effective gas temperature, T_G, varies with spark setting, carburetor-air temperature, exhaust back pressure, and fuel-air ratio; for the present tests these variations, except that with fuel-air ratio, may be neglected. A large number of tests for several cylinders[1, 2] led to a choice of 1150 F for T_G for the head and 600 F for the barrel for a maximum power mixture.

Equations for the average head and barrel temperatures of the Continental A-75 engine were obtained by means of tests varying in turn the pressure drop across the engine and then the indicated horsepower, holding all other engine and cooling conditions constant. The tests were conducted with maximum-power mixture and T_G was assumed to be 1150 F for the head and 600 F for the barrel. The temperature data of Cylinders 1 and 4 were used to determine the average cylinder temperature because there was very little difference in the cooling of the two cylinders. The methods of obtaining the equations are given in Reference 1. The equations are:

$$\frac{T_H - T_A}{T_G - T_H} = 0.031 \frac{I^{0.69}}{(\Delta p\rho/\rho_o)^{0.37}} \quad (3)$$

for the head and

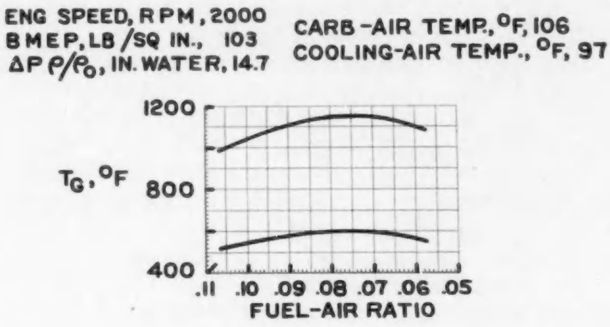
$$\frac{T_B - T_A}{T_G - T_B} = 0.075 \frac{I^{0.66}}{(\Delta p\rho/\rho_o)^{0.51}} \quad (4)$$

for the barrel.

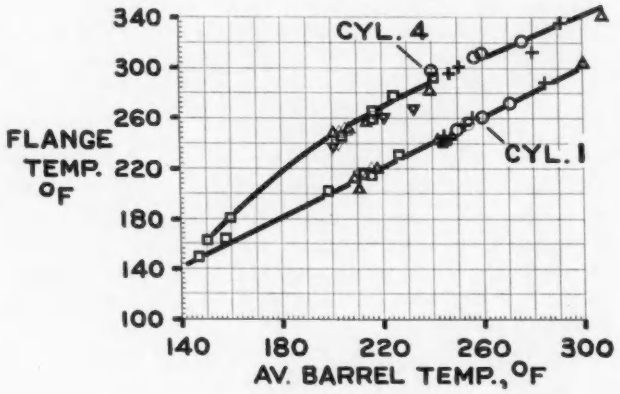
Tests were then made in which all engine and cooling conditions except fuel-air ratio were kept constant. From these tests and Equations (3) and (4), the effective gas temperature for each fuel-air ratio was calculated. The results are shown in Fig. 7 for the head and the barrel.

From the exponents of Equation (3), the data for the

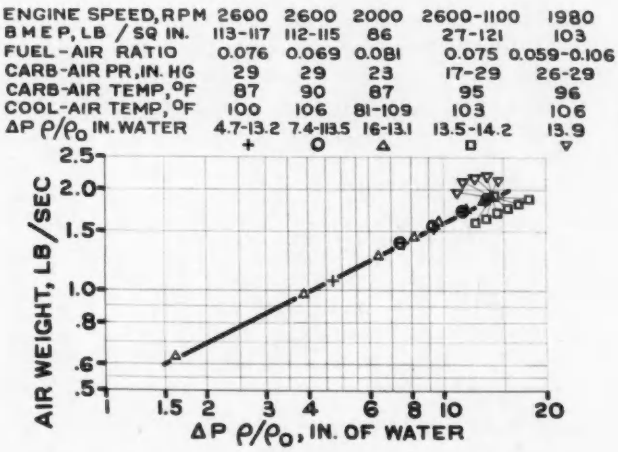
[1] See NACA Technical Report No. 612, 1938: "Heat-Transfer Processes in Air-Cooled Engine Cylinders," by Benjamin Pinkel.
[2] See NACA Technical Report No. 645, 1939: "Correction of Temperatures of Air-Cooled Engine Cylinders for Variation in Engine and Cooling Conditions," by Oscar W. Schey, Benjamin Pinkel, and Herman H. Ellerbrock, Jr.



■ Fig. 7 – Variation of effective gas temperature with fuel-air ratio



■ Fig. 8 – Variation of flange temperature with average barrel temperature



■ Fig. 9 – Variation of air weight with pressure differences

hottest head temperature (on Cylinder 4 at the same location as Thermocouple 14, Fig. 4) were correlated by plotting the left-hand side of Equation (2) against $\Delta p\rho/\rho_0$. The data for the hottest barrel temperature (on Cylinder 4 at the same location as thermocouple 4, Fig. 4) were plotted similarly using the exponents of Equation (4). For both head and barrel temperatures, $F_1(\Delta p\rho/\rho_0)$ in Equation (2) could be replaced by constants (0.055 for the barrel and 0.122 for the head) that agree with former correlations of individual temperature data.

The flange temperatures of Cylinders 1 and 4 are plotted against average barrel temperatures in Fig. 8. The symbols on the curves have the same significance as those shown in Fig. 9. A line drawn to represent a 1:1 ratio of flange temperature to average barrel temperature represents the data for Cylinder 1 accurately enough for all practical purposes. The flange temperature of Cylinder 4 was greater than the average barrel temperature and did not vary directly with the barrel temperature, as shown in Fig. 8. The flange temperature of Cylinder 3 was approximately the same as that of Cylinder 1 and the flange temperature of Cylinder 2 was a little less than that of Cylinder 4. It is therefore possible, from Equation (4) and Fig. 8, to predict the flange temperatures of the cylinders for all engine and cooling conditions.

The quantity of cooling air passing across the cylinders during the final performance and cooling tests is given in Fig. 9 for various pressure differences. All the data taken fall around a single line, as would be expected.

■ Performance and Cooling At Altitude and Sea Level

From the foregoing performance and cooling tests and cooling equations, the performance and cooling characteristics of the engine at an altitude of 8000 ft with maximum-power and maximum-economy mixtures for wide-open throttle and 2600 rpm were determined. The engine is to be operated under such conditions in the projected airplane. The performance and the cooling characteristics at sea level with maximum-economy mixture were also determined. The manifold pressures and temperatures and cooling-air temperatures were assumed to be those of the standard atmosphere.[3] The power at 8000 ft and at sea level for the standard atmospheric conditions was determined from the laboratory tests by the formula that power varies directly as the manifold pressure and inversely as the square root of the absolute manifold temperature. For low altitudes this approximate formula predicts power within a few percent of that predicted by more exact formulas. All calculations were based on cylinder-head temperature never exceeding 500 F.

The brake horsepower at altitude with a cool engine was determined by assuming that the increase in power with a cool engine as compared with a hot engine would be the same as in the laboratory tests. The fuel-air ratio and the indicated specific fuel consumption for maximum-power mixture with wide-open throttle at altitude were assumed to be the same as in the tests with wide-open throttle in the laboratory. A test was made in the laboratory to check this assumption by operating the engine at the same calculated power output obtained with maximum-power mixture at 8000 ft. The indicated fuel consumption for a given

[3] See NACA Technical Report No. 218, 1925: "Standard Atmosphere – Tables and Data," by Walter S. Diehl.

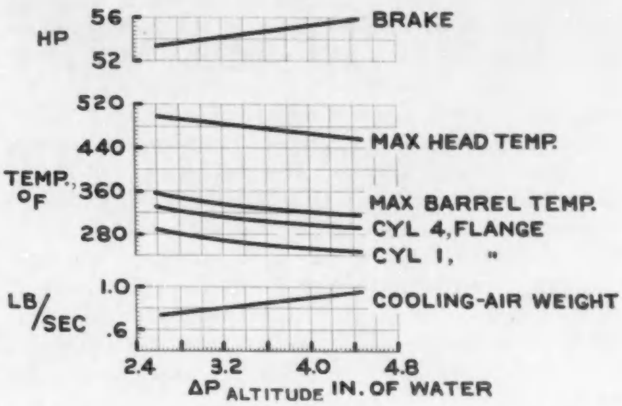
fuel-air ratio under these conditions was practically the same as at wide-open throttle in the laboratory. The tests with a hot and a cool engine showed that the indicated fuel consumption and fuel-air ratio remained constant. The assumption was made that such would be the case at altitude. From the assumption that the mechanical efficiency at 8000 ft was the same as at sea level, and knowing the brake horsepower and the indicated fuel consumption at altitude, it was possible to calculate the brake fuel consumption.

The pressure drop required to cool the engine at the calculated power output at 8000 ft with a hot engine was obtained by means of the equation for the hottest head temperature. This temperature was taken as 500 F, and the cooling-air temperature was obtained from the standard atmosphere tables[3] and the effective gas temperature from Fig. 7. The pressure drop being known, the hottest barrel temperature and the average barrel temperature were calculated by means of the equations previously mentioned, the flange temperatures were obtained from Fig. 8, and the cooling-air weight was obtained from Fig. 9. With the cool engine at altitude, the average cylinder temperature was assumed to decrease the same amount, as compared with the temperatures of the hot engine, as in the laboratory tests. The average cylinder temperature with the cool engine being known, the pressure drop and then the other pertinent temperatures could be obtained by means of the equations and the curves. The performance and the cooling characteristics at 8000 ft with the maximum-economy mixture and at sea level with maximum-power mixture were determined in a similar manner.

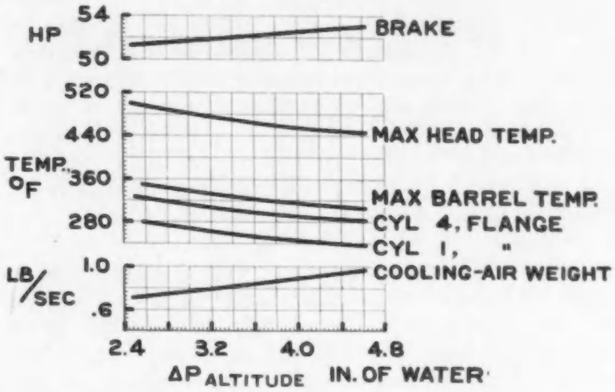
The results of the calculations at 8000 ft with maximum-power mixture are shown in Fig. 10. Only 2.6 in. water pressure drop is required to cool the engine with 500 F as the maximum cylinder temperature when developing about 53.5 bhp. The quantity of cooling air is low enough (about ¾ lb per sec) so that it can be supplied by means of ducts in the wings. The highest barrel temperature is 358 F and the highest flange temperature is 332 F. With about 4 in. of water pressure drop the maximum head, barrel, and flange temperatures can be reduced to 468 F, 323 F, and 300 F, respectively. The quantity will increase to 0.9 lb per sec and the bhp to approximately 55.5.

■ Results with Maximum-Economy Mixture

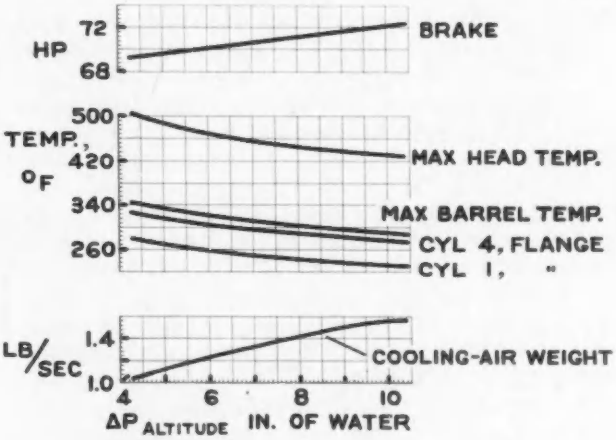
The calculated results at 8000 ft with maximum-economy mixture (Fig. 11) were very similar to those with maximum power mixture. Again, only a very low pressure drop, 2.47 in. water, was required to maintain 500 F as the maximum cylinder temperature. This pressure drop was lower than that required to hold the same temperature with the maximum-power mixture, which seems inconsistent. It would be expected that, with the lower fuel consumption, more air would be required for cooling than with the maximum power mixture. The bhp is lower, approximately 51, with the maximum-economy mixture than with the maximum-power mixture which explains the apparent inconsistency. The hottest barrel and flange temperatures with the maximum-economy mixture are approximately the same as with the maximum power mixture when the highest head temperature is 500 F in both cases. The pressure difference can be increased to 4 in. of water with appreciable decrease in the cylinder temperatures and



■ Fig. 10 – Performance and cooling characteristics of engine at 8000 ft with maximum-power mixture



■ Fig. 11 – Performance and cooling characteristics of engine at 8000 ft with maximum-economy mixture



■ Fig. 12 – Performance and cooling characteristics of engine at sea level with maximum-power mixture

the quantity of cooling air still remains below 1.0 lb per sec. (See Fig. 11.)

More pressure drop is needed for cooling at sea level than at altitude, as shown in Fig. 12. With a maximum-power mixture about 4.5 in. of water is required to maintain a 500 F limit on cylinder temperatures at sea level as compared to 2.6 in. water at 8000 ft. The higher pressure drop required for cooling at sea level would only be needed for a short time at take-off. The bhp at sea level varies from approximately 69 to 72.5. These powers are based on a wet manifold pressure of 29.92 in. hg and, if corrected to a dry manifold pressure of 29.92 in. hg, 75 bhp, the rated power, would be developed with the higher cooling pressure drops. The quantity of air required to maintain 500 F as the maximum cylinder temperature is still comparatively low at sea level being approximately 1.1 lb per sec.

General Discussion

From the foregoing tests the conclusion can be made that the engine cooled satisfactorily with wide-open throttle and maximum-power mixture with 4.5 in. of water-pressure difference and 1.06 lb of cooling air per sec. In addition, from tests on the ground, calculations showed that satisfactory cooling could be obtained at 8000 ft altitude with wide-open throttle and either maximum-power or maximum-economy mixture with 2.6 in. of water-pressure difference and approximately ¾ lb of cooling air per sec. The power required for cooling with the foregoing conditions would be approximately 1.0% of the brake horsepower at sea level and ½% at 8000 ft. The cylinders are small on the Continental engine so that the surface-volume ratio is large, which condition is most suitable for cooling. From this discussion it might be thought that the cooling of engines with large cylinders might be more difficult than with the small Continental engine. Such would be the case if the fin proportions of the large cylinders were about the same as those of the Continental engine. Calculations based on the results of tests[4] on finned cylinders by the NACA indicate that the fin proportions of the Continental cylinders can be changed so that approximately 50% less pressure drop will be required for cooling. The proposed fin proportions can be manufactured with present-day methods. With this improvement in cooling with better finning it is quite probable that the larger cylinder can be satisfactorily cooled with a small pressure difference.

One factor which enters into the cooling of an engine that is submerged in the wing and which is not encountered in present-day installations is the fact that the crankcase is not cooled by air, as previously mentioned. This condition causes the oil to heat up and the flange temperatures to be too high. An external oil cooler of adequate size must be used. In the present tests, it was necessary to hold the oil temperature to 165 F by means of a cooler to obtain satisfactory flange temperatures. The oil cooler could be submerged in the wings, in the same manner as the engine, and wing ducts could be used for cooling.

The shape of the flat engine lends itself readily to the addition of a blower if the latter is needed to obtain satisfactory cooling. With the installation arranged as a pusher, a blower can be placed behind the engine in the propeller support. A blower of the axial-flow type could be easily designed to obtain efficiencies of at least 75%. The propeller support would be larger than if just the propeller shaft passed through it, but could be faired into the wing. In order to use the blower system efficiently, the cooling air after passing over the cylinders could be used to cool the exhaust pipes before exhausting to the atmosphere. Tests, mentioned previously, on the engine showed that the net power, which was the brake power minus the power required by a 100% efficient blower, remained practically constant as the pressure difference increased. If an engine could be cooled satisfactorily with 500 F hottest head temperature without a blower, it would probably be better to use a blower even if its efficiency is 75% and obtain the extra cooling that would prolong the life of the engine. With a blower with 75% efficiency, the net power would be almost as great as without the blower.

One disadvantage of placing the engine in the wing is the impaired engine accessibility. The reduction in operating cost, however, probably will be sufficient to give incentive to efforts to overcome this disadvantage. One point worth noting at this time is that, in the first baffle installation used on the Continental engine in the present tests, the spark plugs were placed under the baffles and no provision was made for removing the plugs without first removing the baffles. In subsequent modifications of the baffles, a hole was cut in each baffle through which the plug projected, the hole being sealed with a leather patch.

A few tests were made on the engine with a standard commercial fuel of 78 octane number with the original intake manifolds to compare the performance of the engine with the performance with 100-octane fuel. Tests were conducted with wide-open throttle, to simulate sea-level conditions, and with part throttle with a manifold pressure equal to the standard pressure at 8000 ft. With wide-open throttle the power decreased approximately 2.5% and with part throttle the power remained the same with 78 octane fuel as compared with power with 100 octane fuel. The decrease with wide-open throttle was possibly due to a lower volatility and lower heating value of the 78-octane fuel as compared with the 100-octane fuel. At part throttle the vaporization of the 78-octane fuel may have been greater, owing to lower manifold pressure, than it was with wide-open throttle, which would increase the volatility and thus cause the power output to be the same as with 100-octane fuel. The use of the lower octane fuel is advantageous in the small airplane owing to its lower cost.

Although there are many problems to be solved in connection with submerged engine installations, the ultimate improvement to be gained more than warrants research on these problems. It was stated at the beginning of this paper that, in some installations, the drag on the nacelles accounted for 25% of the drag of the airplane. Removal of the nacelles from the wings and the use of submerged engines does not mean that the drag of the airplane will be reduced 25%. The wing thickness and taper ratios of modern airplanes, as pointed out by Mead,[5] do not provide sufficient room for the engines. The use of thicker sections, however, with a consequent increase in profile drag plus the cooling and propeller support drag will still probably decrease the drag of small airplanes approximately 15% and the drag of large airplanes about 5% when submerged engines are used instead of nacelles.

[4] See NACA Technical Report No. 676, 1939: "Surface Heat-Transfer Coefficients of Finned Cylinders," by Herman H. Ellerbrock, Jr., and Arnold E. Biermann.

[5] See SAE Transactions, October, 1937, pp. 455-467: "Aircraft Powerplant Trends," by George J. Mead.

The IGNITION SYSTEM as Influenced by FUEL CHARACTERISTICS

by J. T. FITZSIMMONS
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RECENT tests indicate that ignition timing can be used to measure the detonation characteristics of a fuel at various speeds in a multicylinder engine. Present engines usually require a spark to be set below the point of maximum power to avoid objectionable detonation. A more advanced spark timing with a higher octane fuel will give more power where desired.

Unfortunately, the spark-advance requirements with change of speed are not uniform on various commercial fuels from different sources with different processing and blending. Unless a very-high-octane fuel is used, the automatic spark advance on the distributor supplied on the engine may not be equally satisfactory on different fuels. There seems to be no easy or practical way to make possible the adjustment of the ignition timing by the driver to accommodate different fuels now used.

At present it does not seem desirable, from an economic point of view, to increase the accuracy of the ignition distributor if there is much additional cost involved. When fuels become more nearly standardized as to detonation, it may be worthwhile not only to improve the accuracy of the distributor but to change the method by which it is driven by the engine.

ALTHOUGH the spark timing as a control for fuel detonation in automotive engines has been used for a number of years, it is only recently that it has been recognized as an accurate indicator of the detonation characteristics of a fuel. Whereas the ASTM and the Research Method of determining the octane rating of a fuel indicates a value under given fixed conditions, the spark-timing method indicates the detonation characteristics of the fuel over the entire operating range of speed of the automotive engine. By the use of spark-timing adjustment it has been proved that the detonation characteristics of fuels vary with engine speed and that two different fuels having the same octane rating based on the ASTM and Research Methods, may act widely different when used in an engine. This difference is most evident when the characteristics of the fuel are studied over the entire speed range of the engine since the detonation characteristics of some fuels tend to increase with increased engine speeds while, with others, they tend to decrease. It is not my intention to discuss the property of the fuels which brings about this condition since that phase of the subject can be handled far better by engineers more familiar with it, but it does not seem out of place to give a few facts as to the effect which these fuel characteristics have on the performance of the ignition system.

With some engines and fuels combinations, detonation may occur on partly open throttle. Information upon this phase of the problem is rather limited at the present time, and the distributor requirements as related to fuel characteristics will be considered for only full-throttle operation.

Properties of the ignition unit which may affect full-throttle engine performance are accuracy of synchronism and the automatic-advance curve. Accuracy of the distributor synchronism involves variation in timing from cylinder to cylinder from the theoretically correct value. For instance, on a 6-cyl, four-stroke-cycle engine, a spark should occur every 120 crankshaft deg.

The wide-open-throttle automatic-advance curve of the distributor is the advance of the spark timing as related to crankshaft speed, and will depend not only on the characteristics of the engine but on the fuel used. Some of the engine characteristics which may influence the distributor

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 6, 1941.]

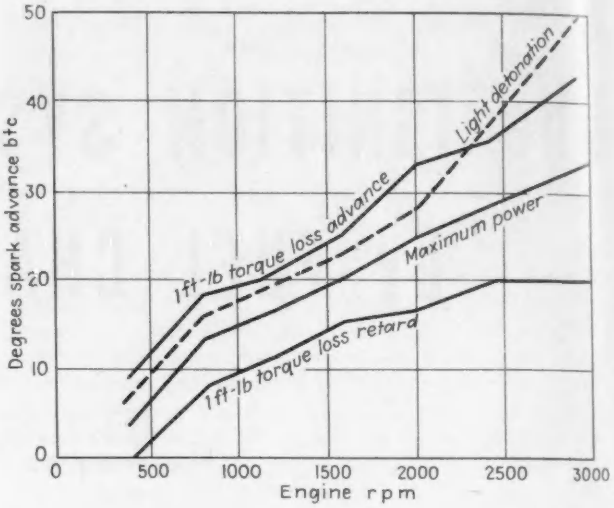
automatic-advance curve are valve timing, fuel mixture ratio, turbulence in cylinder, shape of combustion space, spark-plug location, compression ratio, volumetric efficiency, and the cylinder temperature over the speed range.

An illustration showing how the automatic curve requirements may differ in engines of various designs is shown in Fig. 1. These curves indicate some of the specifications furnished by the engine manufacturers for the automatic curves of their engines, to accommodate which the distributors must be built.

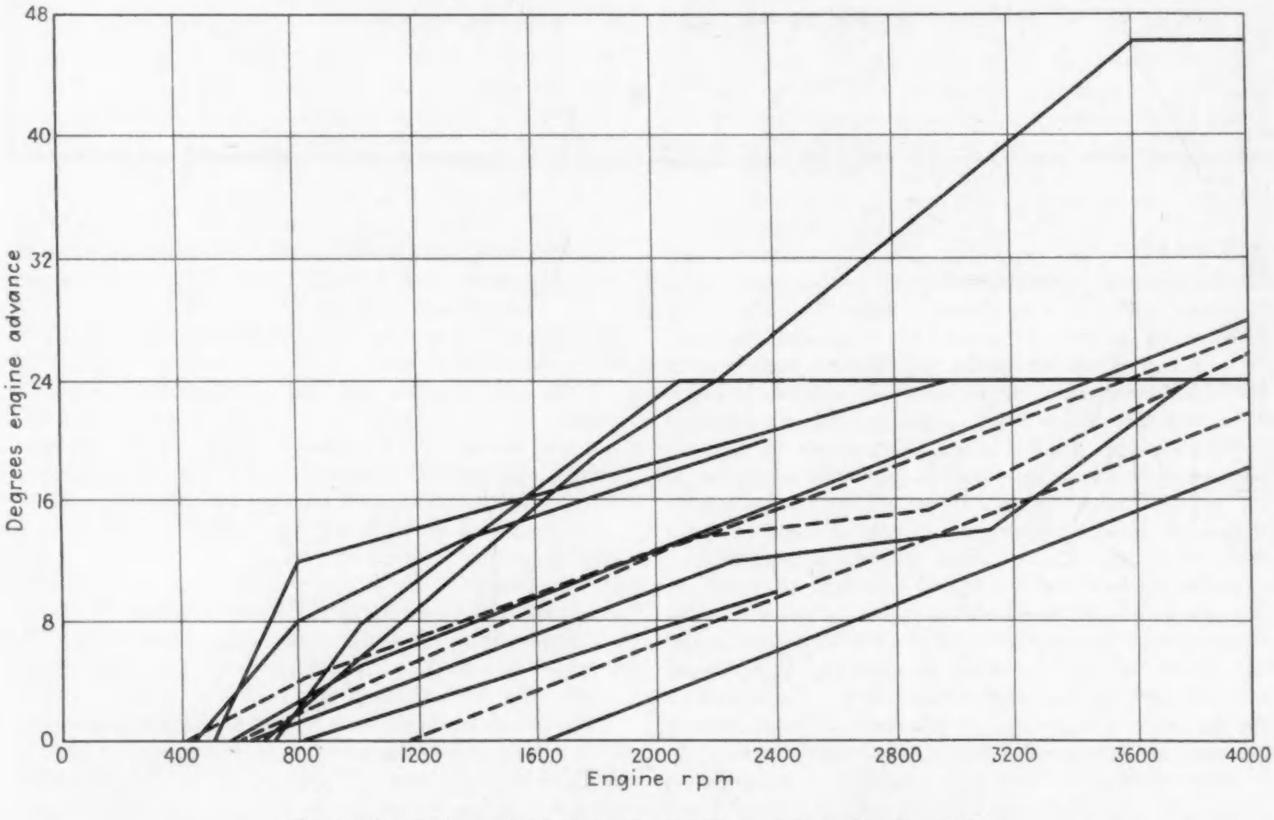
In years past it was customary to furnish a curve which would enable the engine to pull maximum horsepower. In the high-compression engines of today, however, conditions arise which usually require an advance curve somewhat lower than that giving maximum power, if objectionable detonation is to be avoided.

On this point there may be some argument since the compromise curve will be decided by the engineer in view as to what he considers as reasonable detonation. To this result his customer may not always agree. Matters may be complicated further by the use of a different fuel in the field from the one with which the original tests were conducted. If the disagreement becomes too pronounced, the distributor manufacturer becomes involved. If he is building the curve specified by the manufacturer, he is in the delicate position of trying to satisfy the customer without making any change in the curve unless approved by the engine manufacturer.

The use of a few curves as illustration of the general trend in distributor requirements may serve to clarify the discussion. Fig. 2 shows the results of engineering tests on an engine built in 1931 and was furnished us by a manufacturer as a reference for establishing the automatic advance for the distributor. It will be noted that, as is usually the case, the engine was operated at each speed and the spark advance determined which gave maximum power, detonation, and 1 ft-lb loss in torque. The advance which gave detonation is above that which gave maximum power and at no point is the difference between them less than 2 engine deg. The spark advance giving 1 ft-lb loss in torque is at least 5 deg below the point giving maximum power and 7 deg below the point at which detonation



■ Fig. 2 – Results of tests on 1931 engine furnished as a reference for establishing the automatic advance for the distributor



■ Fig. 1 – Automatic spark advance requirements in engines of various designs

occurs. This means that a total variation of 7 engine deg can exist in the synchronism and the automatic advance curve of the distributor, without serious loss of power or appreciable detonation. It also happens that, with the fuel used, the detonation and maximum power curves are parallel over the low-speed range so a slight adjustment of timing can correct the advance, over the entire speed range.

As a further step let us examine Fig. 3, which is another curve on a different engine submitted by the same manufacturer as furnished Fig. 2, but dated in late 1938. Here you will note the curve for maximum power is 1 deg above the detonation curve; consequently, if detonation is to be avoided, it will be necessary to hold the advance curve between that showing 2% loss and that indicating detonation. This operation gives a tolerance of only 3 engine deg on the distributor if detonation is to be avoided and practically maximum power retained. Specifications like this are not difficult to meet, inasmuch as the maximum power and

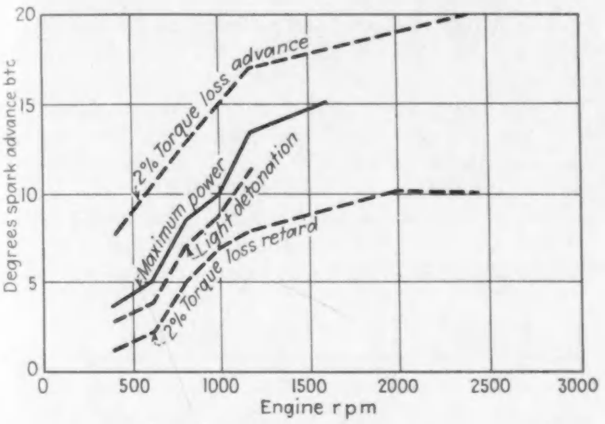


Fig. 3 – Results of tests made in 1938 on different engine than that tested for Fig. 2

detonation curves are parallel and the operator can so set the initial timing of the engine as to have maximum power with slight detonation, or eliminate the detonation entirely at a sacrifice of a little power, over the entire engine speed range. Another alternative would be the use of a slightly higher octane rating fuel with which both maximum power and freedom from detonation could be secured.

The work done by the CFR Road Test Group at San Bernardino confirmed in more detail the information which had been presented to the Society in various papers over the past two or three years – namely that the automatic advance of the distributor should conform to the type of fuels used. Engine manufacturers have probably slighted this part of the work inasmuch as the automatic advance curves of the distributors have been developed for one particular fuel instead of for the complete range of fuels which may be found in different sections of the country. Instead of selecting an automatic advance curve which gives peak performance, it might be better to furnish a compromise curve which, although not giving quite maximum power, would give freedom from detonation with almost any fuel marketed. From service reports we know that this has become quite a problem and, in some sections, service men are changing springs in distributors to satisfy their customers. In most cases investigated, we find that the distributors removed were within factory specifications, but were changed to have lower advance curves to eliminate complaints of excessive spark knock due to fuel or other operating conditions. In almost every case it will be noted that the change was made so as to decrease the advance of the distributor.

Until such a time as fuels may be better standardized as to octane rating over the speed range, it might be well to approach the matter from a compromise point of view, and the results of the tests made on one car with a number of fuels will serve as an illustration. Fig. 4 is one set of the curves taken from the Report of the CFR Committee.[1] Although these curves will differ with different makes of automobiles, they usually follow the same general pattern. The reference fuels indicated by dash lines are not of interest in this discussion. The test fuels as indicated by heavy full or dash lines, representing the spark advance to give borderline detonation, we shall consider as fuels which may be purchased in the open market by the car user. The advance for maximum power, which tests to date indicate does not change with the fuel used, is also shown, as well as the production limits upon the distributor used as standard equipment. As a matter of record, the octane rating of these fuels by the ASTM and Research Method are as indicated in the following, and are also taken from the CFR Report:

Fuel	ASTM	Research
4-B	74.9	73.9
10-B	70.6	83.0
11-B	71.6	78.3
16-B	80.8	81.5
20-B	81.7	95.0

For the manufacturer to establish the proper automatic advance curve for his engine, in view of present information, it would seem necessary that he secure borderline detonation as plotted against spark advance on all fuels which it is anticipated might be used by his customers. There should also be available the spark advance giving maximum power. From these curves a compromise curve should be developed which will lie just below the border-

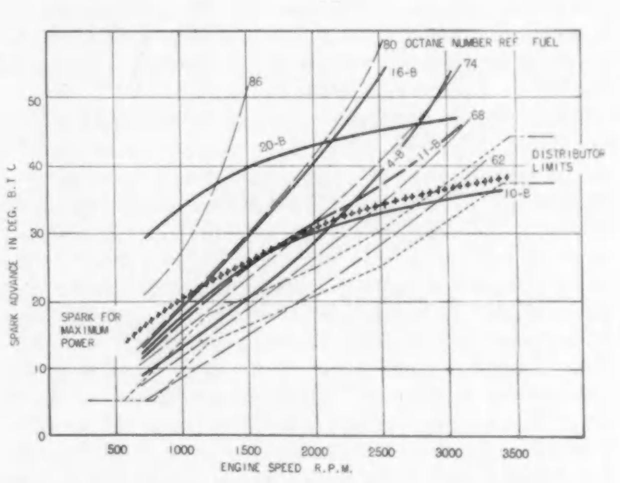
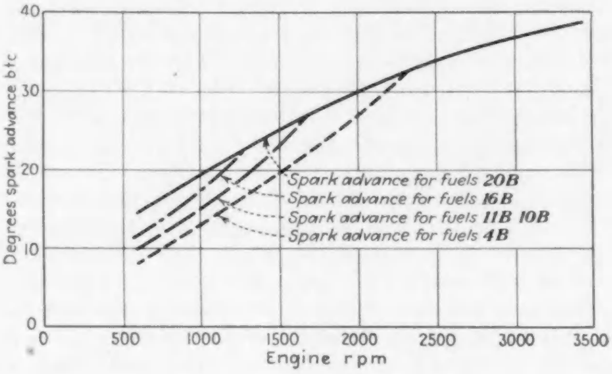


Fig. 4 – Spark-advance curves for 1941 car No. 7 after 8000 miles

[1] See SAE Transactions, May, 1941, pp. 193-204: "1940 Road Detonation Tests (Compiled from Report of the Cooperative Fuel Research Committee)," presented by J. M. Campbell, R. J. Greenshields, and W. M. Holaday.

line detonation curve where this curve is below that giving maximum power with the fuel having the lowest octane rating at each particular speed, but to be on the curve for maximum power where the borderline curve is above that of maximum power.

For illustration, in Fig. 4, considering Fuel 4-B as the one having the lowest octane rating at low speed, it would be necessary to have the automatic spark advance follow



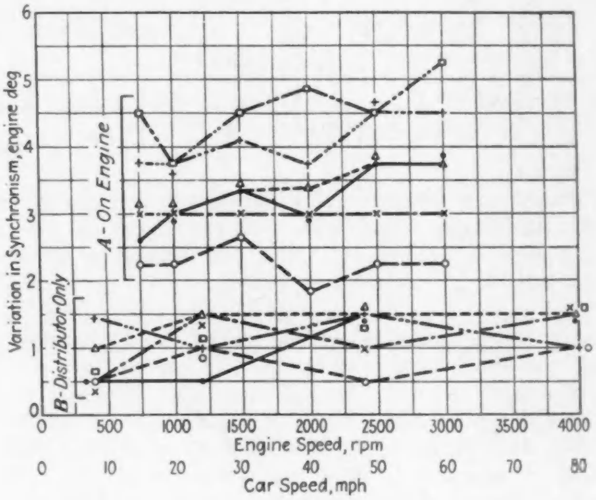
■ **Fig. 5 – Spark-advance curves for fuels 20B, 16B, 11B, 10B, and 4B – 1941 car No. 7 after 8000 miles**

just below the 4-B detonation curve until a speed of approximately 2250 engine rpm is reached, after which it should follow the curve giving maximum power. Considering all the fuels listed, the curves as developed can be illustrated in Fig. 5. Here curve 4 indicates the correct advance for Fuel 4-B. If other fuels were used with this curve, some loss of power would result under 2250 rpm full throttle. If fuel 4-B were omitted from the list, the curve might be made to follow curve 3 with correspondingly less power loss at low speeds while, if 20-B fuel only were used, the automatic curve could be made to give maximum power over the entire engine speed range as indicated in curve 1. Referring again to Fig. 4, note that the production curve now furnished on this engine is below that required for any fuel except 4-B with which some detonation might be expected between 800 and 1700 engine rpm. We do not list all available commercial fuels on this chart, but we do know that, in spite of the fact that it indicates that the commercial timing results in some loss of power, there have been few complaints of either loss of power or detonation coming to our attention so far this year.

The question frequently arises as to the possibility of installing the unit so that a dash control can be used by the driver to adjust the automatic curve for the gasoline he is using. From a theoretical point of view, it would be necessary to change the shape of the curve for the different types of fuels used since the curve changes only at the lower end and in the upper speed ranges follows the full power curve. This consideration would involve quite some complication in distributor construction, but the chief difficulty would be in educating the driver to know when he had the correct setting. Determination of an automatic curve requires quite a little painstaking work for the automotive engineer, and I cannot believe that the average driver can be educated readily to do this. I think that most of us will agree that conditions are very similar to that on carburetors where it has been found advisable to have no easy adjustments except those which affect only slightly the operation of the car.

A few years ago one manufacturer had a dash octane selector, by means of which the initial timing of the distributor could be changed, but it was dropped after a year or two in production. One of the popular priced cars was fitted with a similar device on the distributor itself where the timing could be changed by two knurled screws, but this has also been omitted from current models. The use of some sort of spark control to be actuated by the engine detonation itself offers interesting possibilities but this is also open to objection on some fuels having excessive antiknock characteristics at certain speeds. Here the detonation point is so far above the advance giving maximum power that, not only would power loss result, but danger from broken pistons and blown out-cylinder gaskets, or other types of engine failure might be incurred.

This paper would not be complete without some statement as to improvements in the ignition unit which might assist in the solution of the interrelated fuel-engine problems. It is both desirable and necessary that tolerances on distributor synchronism and automatic advance curves be held to closest commercial limits and in accordance with the limitations of other parts of the equipment. Tolerances in the ignition units have been reduced over the past few years and, at the present time, we are not ready to admit that we are out of step with the industry. The widest commercial tolerance on synchronism at present is 3 engine deg with more than half the units running about half this. It would not be an impossibility to manufacture units with



■ **Fig. 6 – Six distributors tested on a 6-cyl engine**

this closer tolerance if it became justified from an economic point of view. Automatic advance curves are commonly held to 3 or 4 engine deg tolerance. This tolerance would not seem out of reason considering that fuels having the same ASTM rating vary as much as 20 deg in spark requirements over the complete speed range, and engines vary as much as 18 deg between cylinders as a result of variations in compression ratio, temperature, and mixture conditions. As our service organizations are set up today, it is very difficult for even the best to check distributors to within present commercial tolerances. These statements

are not made in criticism but merely to indicate our state of development.

Before closing I would like to bring out a point in regard to the accuracy of timing with respect to the engine, which was made previously in the 1939 Annual Meeting of the Society. This point concerns the type of drive between engine and distributor. This is indicated in Fig. 6, and illustrates the variation in synchronism of a number of distributors at various speeds as checked on the units themselves and after being installed on the engine. It will be noted that a great deal of accuracy in timing between the engine and the distributor is lost in the drive. This loss in accuracy may come from end play in the crankshaft, camshaft, or distributor shaft where spiral or spiral-spur gears are used. With chain-driven camshafts the variation may come from chain slap. Any backlash in gears or the distributor coupling drive adds to the inaccuracies. At some speeds there may be periods when the conditions are much aggravated by torsionals in the engine which cause chain slap, and endwise motion of the camshaft or crankshaft, or a take-up in drive gears and couplings. In some cases the governor of the distributor may hunt, but this condition is not common. In our San Bernardino tests we found engines which, at some speeds, had spark variations as much as 10 deg. Most automatics were locked on these tests, and the conditions in some cases were corrected by a closer-fitting drive coupling on the distributor while, in other cases, there was no easy solution. It has been suggested that the cam controlling the timing of the distributor might be driven directly from the front end of the engine crankshaft. This arrangement involves certain difficulties with the automatic mechanism and requires a separate drive for the distributor of the high-tension system which must run at one-half crankshaft speed. It also places the timing mechanism at the front of the engine where it must be protected from water and dirt. It also adds to the length of the engine, and is rather inaccessible. To date no manufacturer seems interested enough to try it out.

In conclusion, the following suggestions are offered:

Do not hope for a distributor which can be adjusted accurately for various fuels for, even were this possible, the average driver would not know when it was properly adjusted.

Develop the automatic advance curve for each engine which will be a compromise such as to give operation without objectionable detonation where this point is below maximum power, for the poorest fuel which may be used in the engine.

Specify a distributor with close tolerances where necessary and allow them to widen out where not important. You will come nearer to getting a unit which will require less service and be more reasonable in cost. Close tolerances in distributors for most engines are required at the lower speeds and are not so material at higher speeds.

Select the automatic advance curve for satisfactory performance rather than for peak power. Most customers would rather have a car which operates satisfactorily without excessive detonation every day than one with super performance and economy which must be serviced by an expert every time gasoline is bought from a different tank.

Advantages of Compulsory Motor-Vehicle Inspection

FIRST and last, compulsory motor vehicle inspection is a safety project. Its objective is to raise the standard of mechanical condition of the motor vehicle as affects its safe operation and thus aid in reducing the number of deaths and the loss of property resulting from traffic accidents. This is accomplished by virtue of periodical physical tests by trained state or city personnel of the mechanical units of the vehicle that affect its safe operation and by not approving such vehicles until recommended or necessary corrections are made.

We believe it is the duty of public officials everywhere to protect the users of streets and highways in every possible way.

Each state through its motor-vehicle code exacts certain requirements or standards governing motor vehicles and their movement over the highways.

This is in the interest of safety – safety to the operator of a vehicle and safety to other users of public ways.

These requirements follow right along with and are comparable to those imposed upon railroads, steamships, and operators of boilers, elevators, and so on. This is recognized in each instance as necessary for safety and without which pinch-penny and unscrupulous methods of operation and maintenance, if not prevalent, would at least be common and many otherwise avoidable accidents would result.

It is obvious that for any state, to insure its vehicle code being lived up to, must have some system of vehicle inspection and this must be compulsory – compulsory that each and every vehicle be subjected to it.

Without such inspection and where mechanical condition is left up to the individual, it is only in a very small percentage of instances that vehicles are maintained in a safe running condition.

In San Francisco, for example, you will be surprised probably to a point of contradiction, if I tell you that two out of every five cars on the streets today have inefficient or defective brakes and that there are three out of every five that have defects in the lighting equipment, and further that 65% of the vehicles right this minute will not pass state requirements without corrections. This statement is backed up not only by testing station records in Portland but in several other cities and states. It is really something to think about.

A word here might be in order pertaining to the mechanical factors that affect safety.

I have divided these into two groups. In the first group are those we can class as major. This includes: brakes, steering mechanism, and lighting equipment.

In the second group are those regularly required accessories, such as: horn, windshield wiper, muffler, exhaust system, and other miscellaneous items.

Safe operation is dependent upon all these factors. In the first group we believe are those that are most important but, to emphasize one of these over the other, is certainly controversial.

Excerpts from the paper: "Compulsory Motor-Vehicle Inspection," by J. Verne Savage, Superintendent of Municipal Shops, Motor-Vehicle Inspection Station, City of Portland, Ore., presented at the West Coast Transportation & Maintenance Meeting of the Society, San Francisco, Calif., Nov. 6, 1941.

ROLLER BEARINGS and

THE roller bearing has been a peacetime development, particularly since the close of World War I in 1918. Since that time it has been introduced into a multiplicity of industries that now find application on a tremendous scale on defense projects. This paper traces the introduction of tapered roller bearings into all phases of industry, manufacturing, and transportation – particularly the automotive industry, the railroad industry, steel mills, oil industry, and machine tools.

Discussing design principles, the author brings out that one of the fundamental concepts of tapered roller bearing usage is that they must be mounted in pairs and a second is that the bearings

NATIONAL defense depends to a greater extent upon transportation now than at any previous time in our national history. The facility with which military equipment is moved by motor transport, air transport, rail transport, and water transport has been a determining factor in promoting military success in World War II. The outstanding surprise of World War II is the tremendous striking power of the Panzer divisions, and the foundation of this striking power is the efficient combination of tanks and aircraft and, beyond that, gasoline and roller bearings. These two elements of motor transport have grown up together – one associated with the development of power and the second with the transmission of that power.

America is chiefly responsible for the mass production of motor transport, and certainly for the development of the roller bearing, but this development has been along the lines of peacetime use and it remained for the Nazi powers to apply the principle of mass production, taught by America, to military uses.

While America has been slow in the development on a large scale of military motor transport power, it is a foregone conclusion that, having started on this development in a large way, it will, in the course of reasonable time measured in months rather than years, assume world leadership. It is anticipated that the complexion of World War II will gradually undergo a change as America hits its stride in the production of military equipment but, until that time, we must steel ourselves against a succession of unfavorable military events, not to say catastrophies, against the democratic powers.

The present war is not a war of men, for witness the successive collapses of Poland, Norway, Denmark, Belgium, Holland, France, Jugoslavia, Bulgaria, Rumania, and Greece, all of which countries had plenty of men and which, together, certainly outnumbered the German force brought against them. In addition to lack of unified action against the aggressor, they lacked the striking power incident to motorized equipment, backed up as it must be by a tremendous centralized industrial plant for military equipment.

■ Peacetime Expansion Great

The roller bearing has been a peacetime development, particularly since the close of World War I in 1918, and since that time it has been introduced into a multiplicity of industries that now find application on a tremendous scale on defense projects. If the development of the roller bearing had been carried on over the past 23 yr with the thought of national defense uppermost in mind, it could not, during that time, have been more effectively evolved than it has been with peacetime demands in mind. These demands, in our company, have had to do with more and more difficult applications of roller bearings to industrial and transport equipment of all kinds.

[This paper was presented at a meeting of the Washington Section of the Society, Washington, D. C., May 13, 1941.]

There could not have been, from the point of view of national defense, a more fortuitous event than the collapse of the automobile industry in 1920. This event illustrated in a most emphatic manner the futility of dependence upon a single industry as the outlet for a highly specialized product such as the roller bearing. The collapse of the automobile industry to 10% of normal in 1920 led to a complete reorganization of the company's activities, and an intensive drive was made on the development and introduction of roller bearings into practically all phases of industrial, manufacturing, and transportation activities. Among the particular lines developed at this time were the railroad industry, the steel-mill industry, the oil-country industry, paper-mill industry, power-transmitting machinery, machine-tool industry, truck-trailer transport, marine equipment and, finally, aircraft in which we proved ourselves capable of making more efficient that highly valuable military branch of the services.

War is a last resort activity of mankind in which is utilized all branches of human activity, and thus the intensive development of the roller bearing in the last quarter century and its application in practically all mechanical fields of human endeavor can be considered as a significant contribution to the forces of the opposing sides in this present crisis of the world's history.

Before proceeding with a detailed description of the application of roller bearings to various branches of military science, it is necessary to say something in reference to the design, selection, and use of the roller bearing. These phases of roller-bearing use are described in complete detail in the *Timken Engineering Journal,* the detail of the design being covered in the first 12 pages, bearing selection in the next 30, and bearing lubrication on pp. 348-356, inclusive.

The tapered-roller bearing, the specific subject of this article, is built upon the principle that the rolling elements are tapered to meet on a common point on the axis of the bearing. This design introduces certain fundamental con-

NATIONAL DEFENSE

accommodate any combination of thrust and load; for high thrust reactions, the bearing is made with a steep taper. He points out that tapered roller bearings are rated in accordance with a speed factor, life factor, and application factor, explaining how each of these factors is derived.

In the remainder of the paper are discussed: lubrication, extreme-pressure lubricants, lubricant testing, types of tapered roller bearings, contact stresses in solid and hollow rollers, starting friction on railroad axles, crankshaft and crankpin application, machine-tool applications, steel rolling mill applications, and oil-well applications.

by T. V. BUCKWALTER

Vice President, The Timken Roller Bearing Co.

cepts of bearing usage, the principles of which are, first, that the bearings must be mounted in pairs, or built so as to function in pairs and, second, that the pair of bearings accommodates any combination of thrust and radial load. The angle of taper is generally selected so that the thrust and radial capacities are approximately equal, it being kept in mind that all of the rolls function in thrust, whereas a relatively small percentage, generally considered as 20%, function in carrying radial loads. The included angle of the bearings is therefore in the general neighborhood of 20 to 40 deg, depending on the character of the service. The tapered-roller bearing consists of two units, first the cup or outer race and, second, the cone comprising the inner race and roller assembly. This combination, at first sight, may appear a disadvantage from the point of view of commercial assembly, but actually it is one of the outstanding advantages of the tapered-roller bearing due to the fact that the cups can be mounted in the wheel hub or machined housings as separately manufactured parts – which means that the separate parts may be made in widely separated plants – and this is of strategic importance in wartime. The cones are mounted on the shafts which are likewise separately manufactured parts, and generally the first contact between the mating cups and cone is on the assembly operation, often in progressive-assembly lines. It is necessary to figure complete interchangeability to the highest degree of accuracy, and to manufacture accordingly, in order to acquire and maintain this facility of assembly. In our company this work has been carried on through one of the most complete assemblages of manufacturing, inspection, and gaging machinery available in America and, it may be said, in the world. Rolls are held within minute dimensions as to accuracy of taper, roundness, and size, and an equally accurate system of manufacturing holds the cones and cups to narrow limits on finish, taper, and size. The taper principle develops an inherent individual thrust reaction on the part of the taper rolls, and this reaction is resisted by an integral rib built on the cone. Advantage is taken of this inherent thrust of the individual rolls to utilize the square end of the roll to develop a self-squaring action on account of the appreciable surface contact between the end of the roll and the cone rib. This action improves bearing performance, reduces friction, and increases the capacity of the bearing to withstand continuous loads as well as shock loads.

The taper-roller bearing is made in single-row type with normal taper and also with steep taper for the purpose of taking high thrust reactions. The cups are made plain, but cups with integral rib to facilitate assembly in machine tools, wheels, and other mechanisms are available as standard. Bearings are made in two-row type with two single cones and double cup, and in two-row type with double cone and two single cups. Further combination is to make the bearing in four rows, the so-called "quad," generally consisting of two double cones, one double cup, and two single cups. Taper-roller bearings are also made for pure thrust service and, as such, are developed for a wide variety of loads, bearings varying from those used for the steering pivot of automobiles to those carrying the heavy thrust loads of thousands of feet of drill pipe as used in the oil service.

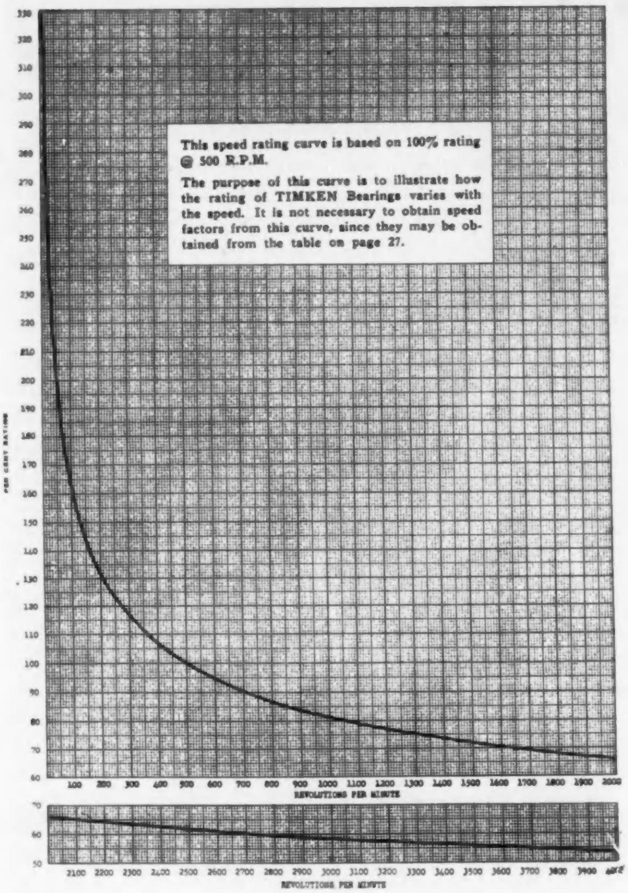
Bearing Ratings

Bearing rating has been a subject of intensive study over a long period of years, and this study is correlated with a long experience in building 75,000,000 taper-roller bearings annually with servicing facilities. The ratings of the bearings are shown in the catalog and this is built around the speed-rating curve shown in Fig. 1. Normal is 100% at 500 rpm. This becomes 200% at 50 rpm and, at 60%, the speed can be equivalent to 2700 rpm. Bearing selection is best handled by a group of engineers specializing in that work. They have the experience of thousands of applications at hand, and have a wide experience in correlating "speed factor" and "life factor" based on the expected life, and "application factor" depending on the character of loading, the variety of shocks sustained and a "combined factor" taking all these things into consideration. However, where because of time or for other reasons it is not desirable to contact this group of engineers, a bearing selection can be made by careful perusal of the data shown in the *Timken Engineering Journal.* A life-expectancy curve is indicated in Fig. 2. This is based on experience whereby at 3000 hr a failure of not over 10% of the bearings is developed which is considered a life factor of 1. The life factor of 2 will develop 30,000 hr, and a life factor of ¾ only 1200 hr. The life factor should be selected so that the bearing life is generally equal to the life of the machine in which it is applied.

Extreme high speed equipment, such as grinding machinery, is lubricated by the cannibal system whereby minute volumes of oil are fed to the bearing drop by drop, either by needle valve or through the medium of a wick. The rolls should not be allowed to dip appreciably into the oil as the churning will develop considerable heat. Bearings working under extremely high duty such as railroad bearings, particularly for a locomotive on driver axles, are best lubricated by a medium heavy oil of good to high quality. The railroad grade of oil known as "superheat valve oil" is excellent for heavy bearing lubrication. The slight amount of animal matter in valve oils improves the "oiliness" properties of lubricants, serves to protect the bearing surfaces from corrosion, is less likely to loss through enclosures, and is in general a satisfactory lubricant for heavy-duty applications. Railroad car oil is not so desirable on account of the higher content of acid therein which has a tendency to attack bearings which, due to service conditions, may be left standing for considerable periods of time. Where greases are indicated for bearing lubrication, it is important to obtain greases of good quality. A grease lubricant is never any better than the straight oil from which it is based. If the oil is of high quality, it is generally not necessary to use more than 8-10% soap. Unfortunately, many of the greases on the market consist of about 70% red oil and 30% soap. These greases are satisfactory for light service but, under no circumstances, should they be used under continuous or heavy service where a long period of bearing life is expected.

Speed Rating Curve

The upper section of the curve indicates the per cent rating up to 2000 revolutions per minute. The lower section gives the per cent rating from 2000 to 4000 revolutions per minute.



■ Fig. 1

Life Expectancy Curve

Hours Expectancy	Life Factor f	Hours Expectancy	Life Factor f	Hours Expectancy	Life Factor f	Hours Expectancy	Life Factor f	Hours Expectancy	Life Factor f	Hours Expectancy	Life Factor f
500	.583	1500	.812	2500	.947	5500	1.200	12500	1.537	50000	2.333
600	.616	1600	.828	2600	.958	6000	1.233	15000	1.622	55000	2.400
700	.645	1700	.843	2700	.969	6500	1.263	17500	1.700	60000	2.458
800	.672	1800	.857	2800	.979	7000	1.291	20000	1.770	65000	2.524
900	.696	1900	.872	2900	.990	7500	1.318	22500	1.834	70000	2.581
1000	.718	2000	.885	3000	1.000	8000	1.343	25000	1.893	75000	2.635
1100	.739	2100	.898	3500	1.047	8500	1.368	30000	2.000	80000	2.687
1200	.759	2200	.911	4000	1.090	9000	1.392	35000	2.093	85000	2.738
1300	.777	2300	.923	4500	1.130	9500	1.415	40000	2.181	90000	2.784
1400	.795	2400	.935	5000	1.166	10000	1.437	45000	2.259	95000	2.829

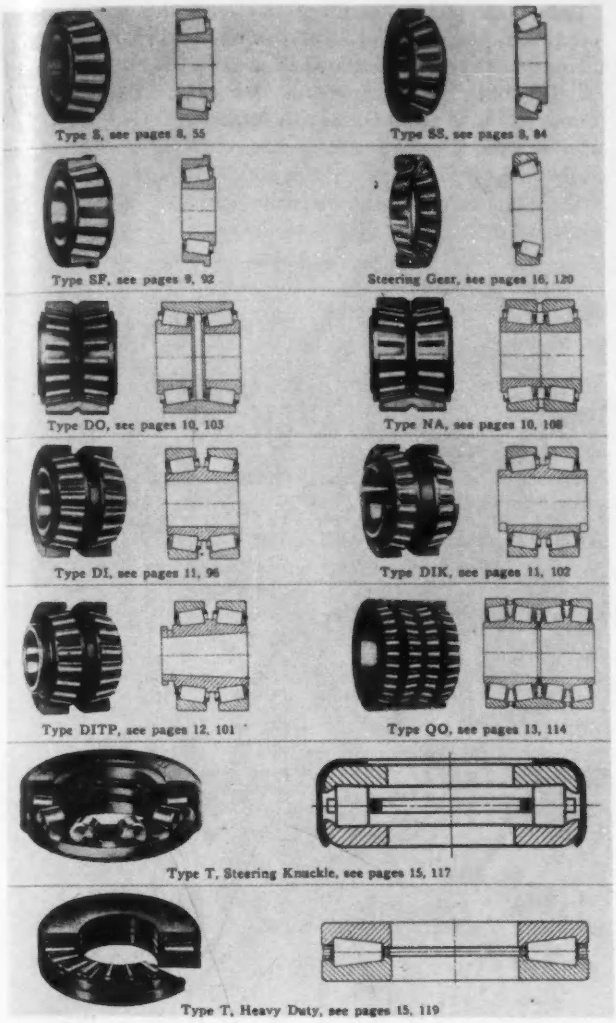
■ Fig. 2

■ Lubrication

Lubrication is an important item in the successful use of roller bearings. A lubricant is essential to allow for the minute inaccuracies arising in the manufacture of the bearings. While these inaccuracies are minute, measured in diameter by 0.0001 in. and in finish of the order of 10 micro-in., they are nevertheless sufficient, and in a bearing not lubricated or in a dry state, they cause metal pick-up and gradual attrition of the bearing surfaces. The second function of the lubricant is to protect bearing surfaces from corrosion. Another function is to dissipate heat and prevent abnormal rise in temperature, having in mind the great carrying capacity of a roller bearing within a limited compass. An important function of the lubricant enclosure is to keep out water, dirt, acid, and abrasives, the entrance of which would have a deleterious effect on the bearings and, at the same time, we must retain the lubricant and prevent any undue loss.

A grease is used in ordinary bearing service. This service applies to automotive front and rear wheels, differential housings with bevel gears, and transmission. A grease is generally indicated on similar industrial applications.

The use of bearings at high speed indicates the need for an oil. Extremely high speeds require lighter oils of lower viscosity for the reason that the churning of the oil at high speeds becomes an important factor in generating heat.



■ Fig. 3 – Timken bearings, single-row, multiple-row, and thrust

Specifications for lubricants and types of lubricant indicated for variety of services are shown in the *Timken Engineering Journal*, pp. 349-352, inclusive.

■ Extreme-Pressure Lubricants

The development of the roller bearing for steel-mill applications, involving loads measured in millions of pounds per bearing, and where the load under certain circumstances approaches 100,000 lb per lineal inch of roller contact, developed early in our experiments the need for a lubricant having much greater film strength and, therefore, carrying capacity than have straight mineral oils. Our company took the initiative in the development of such a lubricant and carried on a long series of experiments in introducing various metalloids into the highest grade of straight mineral oils. A satisfactory lubricant was developed in which the carrying capacity of the lubricant measured in film strength on a lubricant-testing machine is normally four to six times that of the straight mineral oil with which it is compounded. The "EP" so-called extreme-pressure lubricants are now available from a number of sources and their use is indicated where extreme heavy overloads are carried in bearing service.

■ Lubricant Tester

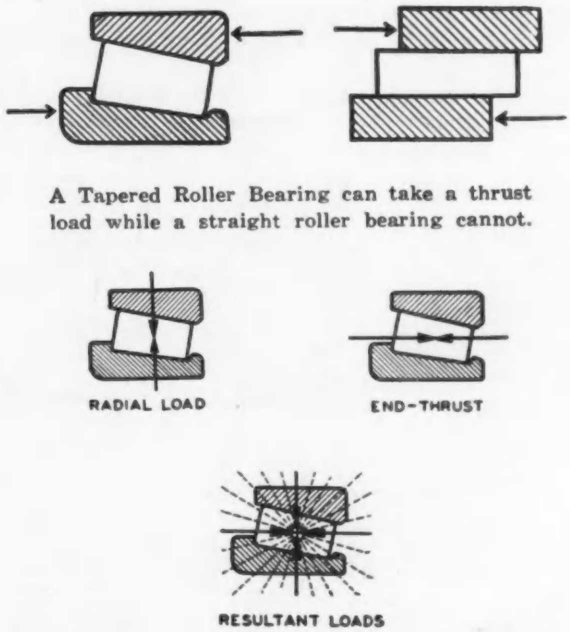
A machine for testing lubricants was developed by the company for the primary purpose of assigning definite numbers to the film strength or oiliness of lubricants. This machine indicated that an ordinary inexpensive red oil has a film strength of about 3000 psi. The better grades of straight mineral oils have 8000 to 10,000 psi. The application of "EP" elements to these good grades of straight mineral oils increases the film strength to the neighborhood of 25,000 psi.

Long-life bearing experiments, in heavy service, such as steel-mill roll necks, railroad bearings in locomotive drivers, with superior lubricants, indicate that the lubricant film does not break down between the roller and race contact and that the wave of lubricant preceding and following the roller contact serves greatly to increase the effective area of this contact, and reduces correspondingly the stresses tending to limit the fatigue life of the roller-bearing elements.

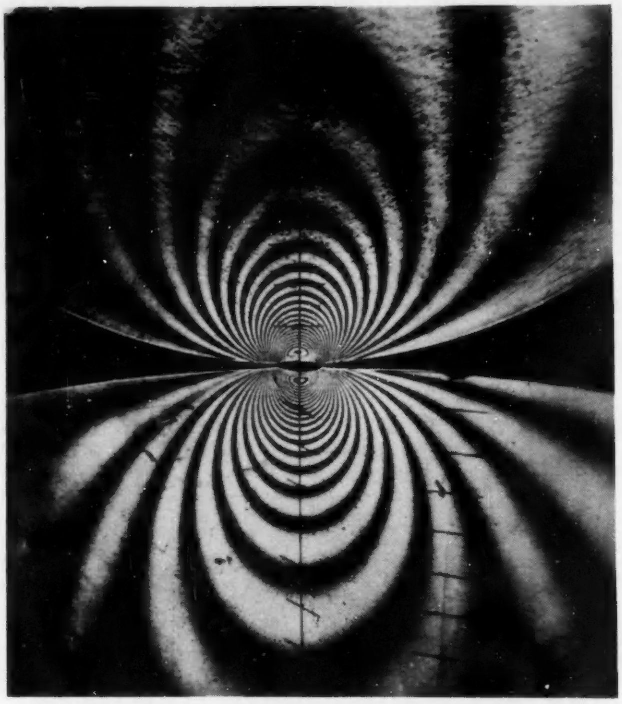
■ Applications to Military Machines

Fig. 3 shows the standard forms of taper roller bearings. The top row shows the standard bearing and the steep-angle bearing. The second row shows the ribbed cup and the cup-and-roller assembly for operation directly on a raceway formed on the shaft. The next row shows the double bearings with double cup, adjustable and non-adjustable. The fourth row shows the double bearing with double cone. The fifth row shows the double bearing with taper bore and also the quad. The thrust bearings show the self-contained type for light- and medium-duty and the fully-ground type for heavy duty.

Fig. 4 illustrates in diagrammatic form the principles under which the taper roller bearing takes any combination of thrust and radial loads. A straight bearing is illus-



■ Fig. 4 – Principles under which the taper roller bearing takes any combination of thrust and radial loads



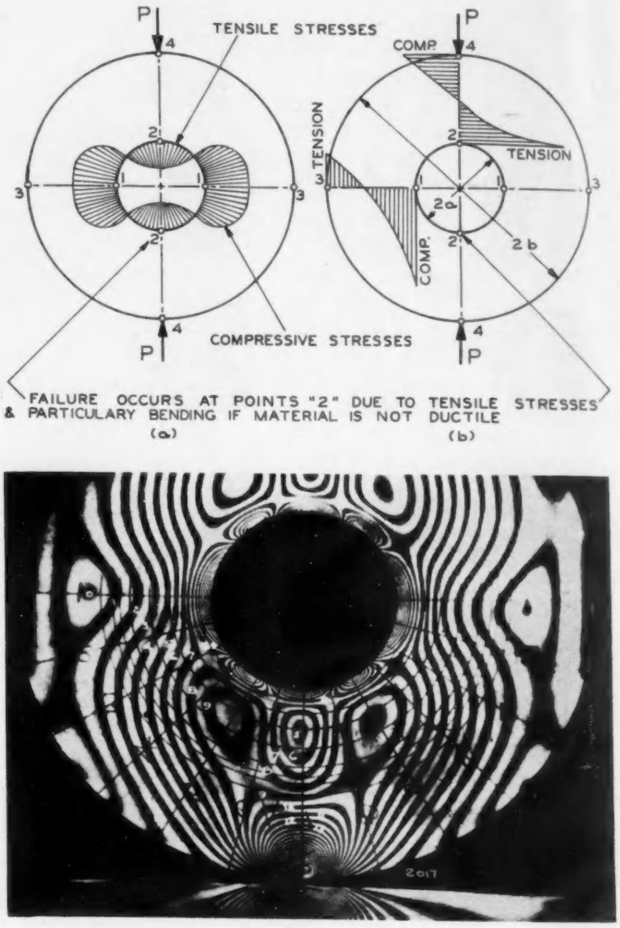
▪ Fig. 5 – Photoelastic diagram of stresses set up at point of contact between roller and race

trated as a matter of comparison and which requires an independent bearing for taking thrust reactions.

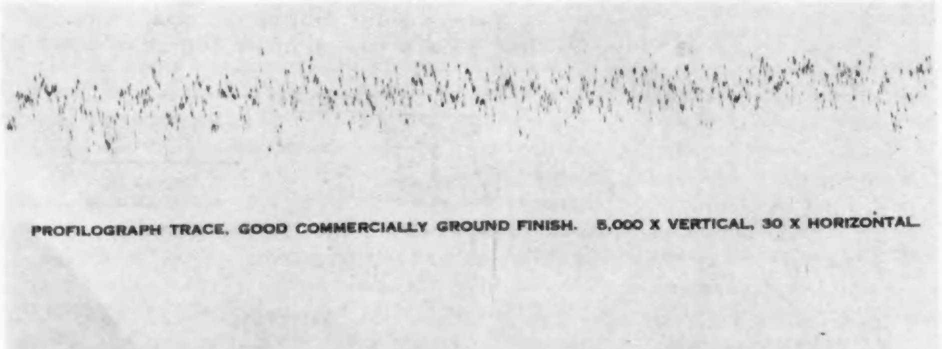
Roller-Bearing Contact Stresses – Fig. 5 is a photoelastic diagram showing the character of the stresses set up at the point of contact between roller and race. The magnitude of the stresses is measured by the number of stress fringes. The highest stress occurs in a zone materially under the surface of the rolling elements. The treatment of the steel requires the greatest resistance to distortion and flaking in the zones of highest stress. A gradual transition from hard case to tough core is derived from the heat treatment. The highest stress concentration is located well within the case.

Stresses in hollow rollers are illustrated in Fig. 6. The hollow roller has material advantages in bearing assembly in the larger sizes, but the character of the stresses set up on the surface of the roll and in the surfaces of the bore delimit the relative proportions of the bore to the diameter of the roll. Extreme care in this respect in the design removes the limitations against the use of hollow rollers and bearing life dependent upon fatigue of the rolling surfaces is obtainable with properly designed hollow rollers.

The character of the finish of the best roller-bearing



▪ Fig. 6 (above) – Stresses in hollow cylinders – Lower illustration shows fringe photograph of hollow cylinder



▪ Fig. 7 – Finish of best roller-bearing practice compared with conventional ground finish

practice, termed "mirror finish" as compared with conventional ground finish, is illustrated in Fig. 7. Vertical magnification is 5000. The error in the mirror finish is limited to 6-10 micro-in., whereas a commercially ground finish is many times greater as indicated.

Ratio of bearing life to hardness is illustrated in Fig. 8. The full hardness bearing of 627 Brinell has a life 2200 times greater than an unhardened bearing of 240 Brinell at 100% rating. The relationship at 75% rating load generally used is 5000:1.

Roller-Bearing Friction – The relative starting friction of the roller bearing and plain bearing on the basis of the railroad axle at various loadings from 3000 to 25,000 lb per journal is indicated in Fig. 9. The ratio is about 30 to 1 on the loaded bearing in favor of the roller bearing. This ratio reduces immediately with rotation of the plain bearing, and under the best operating conditions at normal operating temperature the roller bearing resistance is about 10% less than the plain bearing.

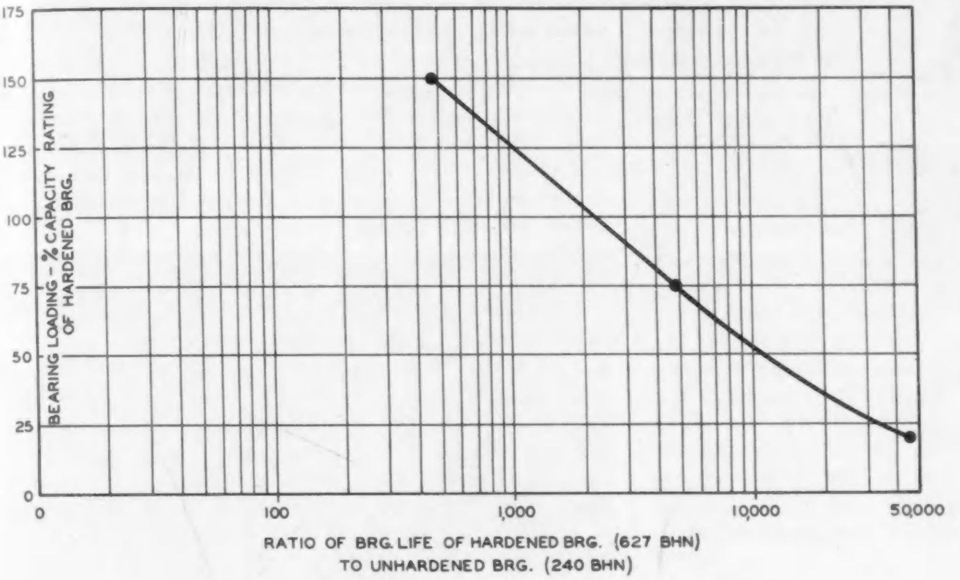
Roller bearings in automobiles include the application to front and rear wheels, differential, pinion, transmission, and steering gear. The applications on military craft are generally similar to commercial automobile practice.

The adaptability of a pair of roller bearings to take any combination of radial and thrust load is beautifully illustrated in the automotive front-wheel and steering-pivot assembly. The general practice is to mount the load almost over the inner bearing, which relieves the outer bearing of much of the radial load, but which imposes full thrust reaction thereon. The steering pivot bearing has the great advantage of maintaining uniform friction and resistance to starting under practically all conditions of lubrication. The relative size of the thrust bearing is small compared with the radial bearing.

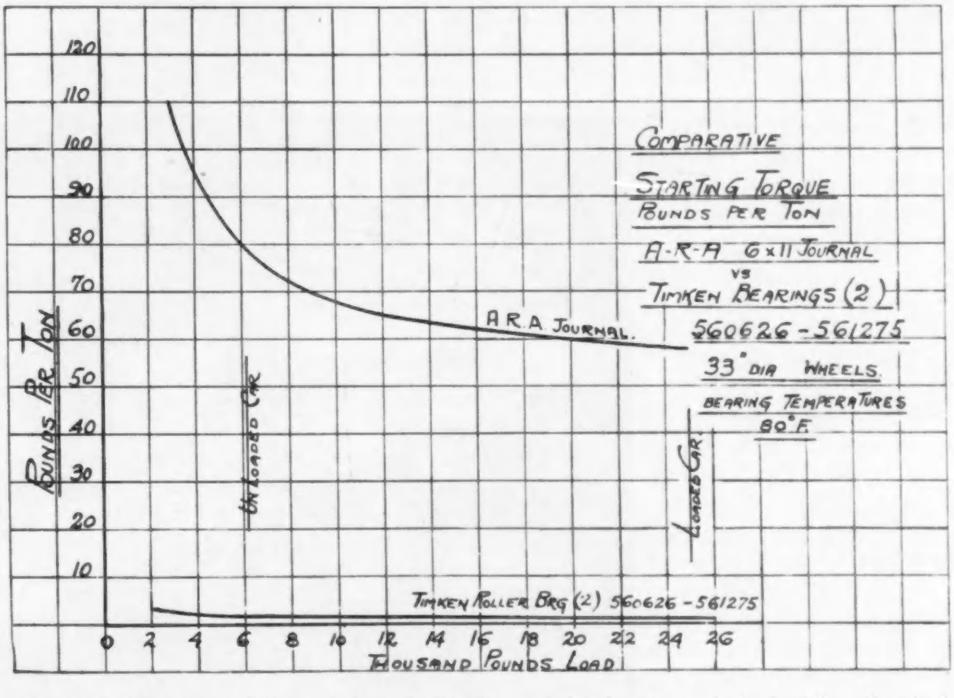
Single-bearing application for rear wheel is the simplest application with taper-bore bearing mounted on a taper shaft continuous with the hub seat. The application requires butting of the shafts at the center to provide thrust stability in both directions. It is a successful and widely used rear-axle mounting.

In the double bearing for rear wheel, the complete combination of thrust and radial loads is taken on the pair of bearings supporting each wheel. The construction is adapted to the heavier automotive and military applications. The necessity of butting axle shafts together at the center is eliminated.

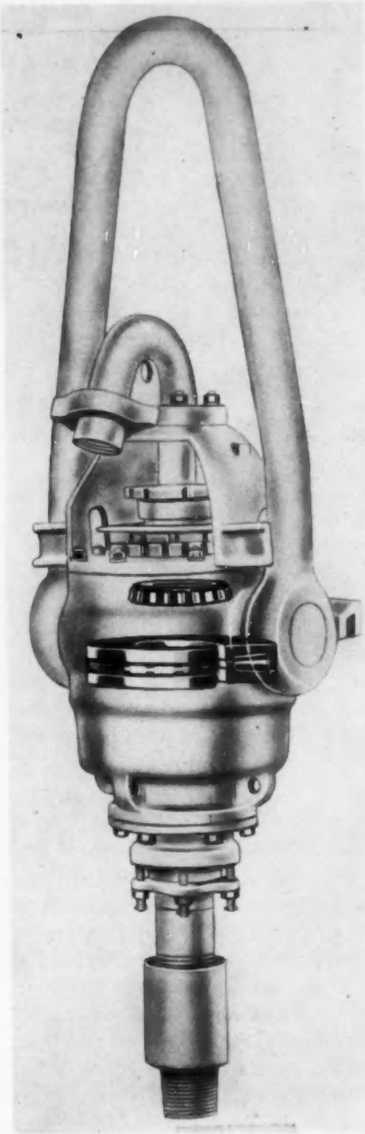
In the floating rear-axle assembly the axle shaft is relieved of radial load which is transmitted directly to the tubular axle housing. The stability of the mounting is reinforced by the axle shaft. The construction is well adapted



■ Fig. 8 – Ratio of bearing life to hardness



■ Fig. 9 – Relative starting friction of the roller bearing and plain bearing on the basis of the railroad axle



■ Fig. 10 – Application of full thrust bearing to oil swivel

for heavy axle applications.

In the straddle pinion mounting the thrust and radial loads are carried on the double taper-roller bearing. A third bearing of the straight roller type supports the overhang of the pinion. The construction has extensive application in automotive vehicles.

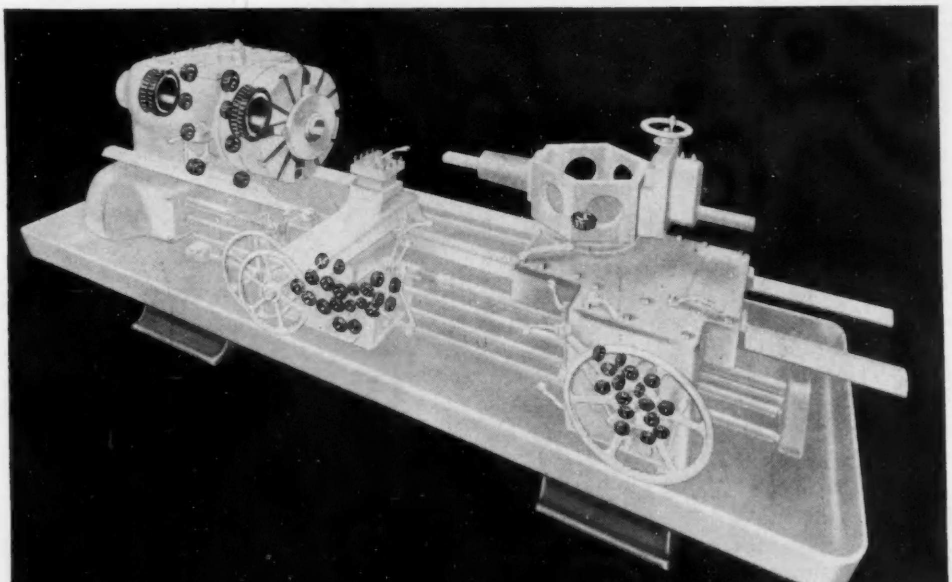
With the overhung pinion and differential mounting, the pinion is overhung regarding the taper bearing but the appreciable bearing spread holds the gear and shaft deflection within approved safe life limits. The differential mounting is of the non-adjustable type, requiring close tolerance in bearing manufacture and equally close precision in differential hub and housing manufacture. The application is a very successful automotive drive for commercial and military use.

In the worm drive and gear the so-called direct worm-gear mounting is employed with thrust reactions taken on the forward worm-shaft bearing. It is a simple mounting, highly successful. The differential is of the conventional adjustable construction, well adapted to heavy rugged commercial and military service. An alternate design is to mount the worm-shaft bearing in the opposite direction, taking the thrust reactions on the rear bearing.

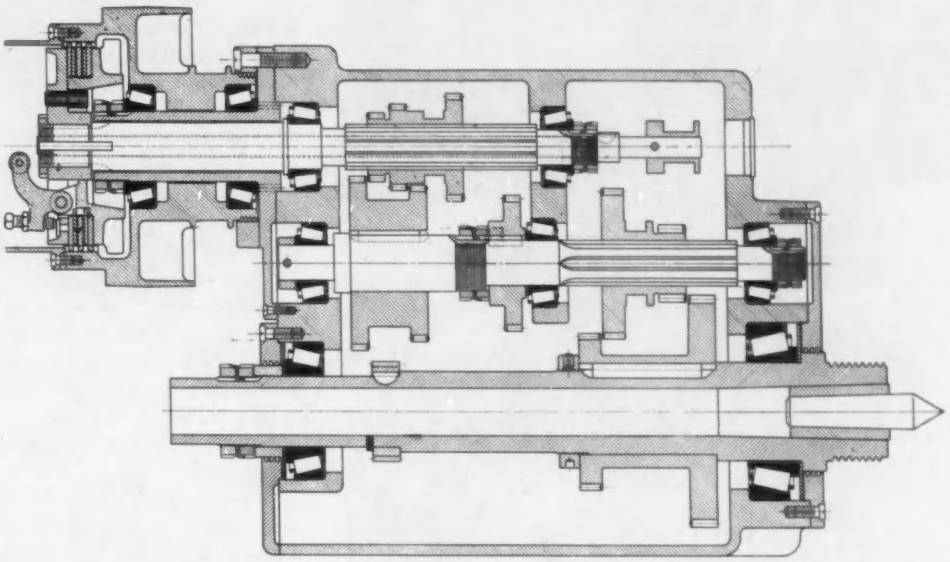
The oil well swivel shown in Fig. 10 illustrates the application of the full-thrust bearing which carries tremendous thrust loads in supporting the drill pipe requisite for the drilling of the deepest oil wells. A radial bearing steadies the swivel and carries the weight of the swivel under certain loading conditions.

In the case of crankshaft and crankpin application, a double bearing supports each end of the crankshaft and a similar bearing on the crankpin is mounted in the large end of the connecting rod. The application shows the adaptability of the roller bearing for taking high loads subject to wide variation in extent and operating under severe service conditions.

The roller bearing application to a lathe is illustrated in Fig. 11. The principles as applied to the transmission of power in the automobile apply to the spindle drive and feed mechanism bearings on machine tools. The tremendously increased capacity of the modern machine tool is due primarily to the preloaded condition of the main spindle bearings. These bearings are adjusted neat with zero lateral freedom and are then pulled up with about 0.005 in. tighter adjustment. This adjustment forces all the rolling elements into a condition of compression, the compressive force being such that the maximum capacity of the machine tool can be developed without a condition of backlash developing in the spindle bearings. This removes chatter from the machine-tool spindle entirely for the reason that there is no space for the development of chatter because, in addition to the pre-loaded adjustment of the bearing, the cone is tightly fitted on the spindle and the cup mounted snugly in the housing. The productive

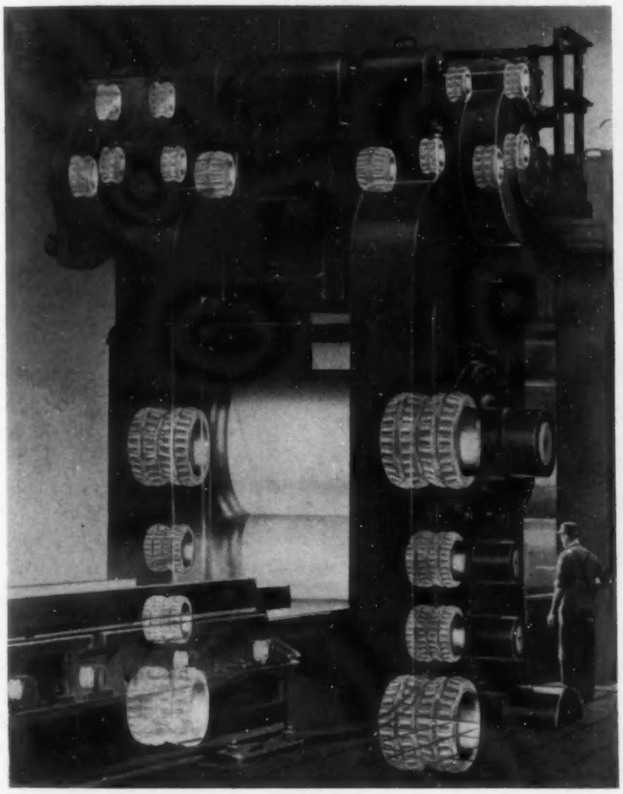


■ Fig. 11 – Application points of tapered roller bearings in a modern turret lathe



■ Fig. 12 – Flanged spindle mounting – lathe headstock

■ Fig. 13 (below) – Application points of tapered roller bearings in a modern four-high steel rolling mill stand



capacity of a machine tool is, in many cases, doubled by the freedom of chatter arising from the preloaded roller bearing. The machine-tool industry has been revolutionized completely by the application of the roller bearing with the consequent reduction in cost of work, improved quality of finish, and greatly extended life of not only the main spindle, but of the entire machine tool as compared with the former plain-bearing mechanism. The capacity of the machine-tool equipment of America has been doubled by this development.

The principle of the pre-loading applies with equal force whether the bearings are mounted singly or in pairs. The drive gearing affords a multiplicity of feeds and speeds essential in the modern machine tool for handling every variety of work required by modern production conditions.

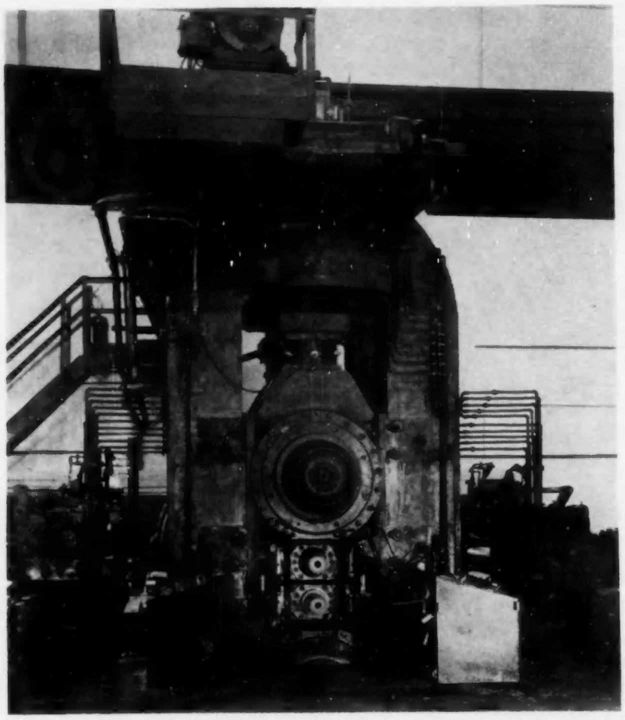
The flanged spindle mounting is shown in Fig. 12. The simplicity and accuracy of the mounting is apparent. This is a typical modern engine lathe drive.

A steel rolling mill is shown in Fig. 13 and illustrates a modern four-high mill with the large massive back-up rolls supporting the smaller work rolls. The loads imposed on bearings of this character are measured by millions of pounds. Space limitations require the use of the quad bearing to develop the tremendous carrying capacity within the limits imposed by mill construction. Bearing applications for the control mechanism and screw-downs are indicated. The steel-mill industry, particularly with reference to production of plate sheet, bars and tubes has been revolutionized completely following the introduction of the roller bearings. The continuous plate mill has been made a commercial possibility by the roller bearing as the guiding of the relatively thin plates is impossible by edging and is accomplished by regulation of the screw-downs. The bearing wear is a negligible matter after the rolling of hundreds of thousands of tons of steel. The steel-mill bearing application is a military asset of the greatest value in the current emergency, requiring tremendously increased tonnages from our steel industry.

An end view of a roller-bearing four-high steel mill is illustrated in Fig. 14. The proportions of the assembly can be visualized from the bore of the back-up roll which is approximately 32 in.

The relative size of a back-up roll and the chucks mounting the bearing can be appreciated further from a study of Fig. 15. There is no country in the world as well equipped for the economic production of steel in volume and accuracy as the United States, the industry having been completely revolutionized by the introduction of the roller bearing.

A roll-neck mounting is shown in Fig. 16. This is illustrative of the steel-mill mounting but is equally applicable on military work, particularly the trunnions of heavy ordnance. The bearings have stood up for years in continuous

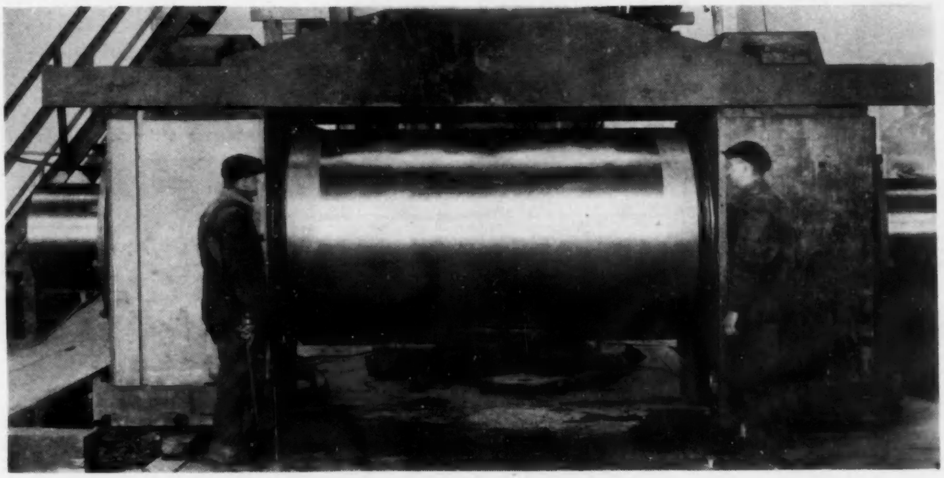


duty under loads measured in millions of pounds. The bearings can be set up for trunnion mounting in a preloaded condition, removing bearing back lash entirely even under heavy firing reactions. The life under proper lubrication should be infinite.

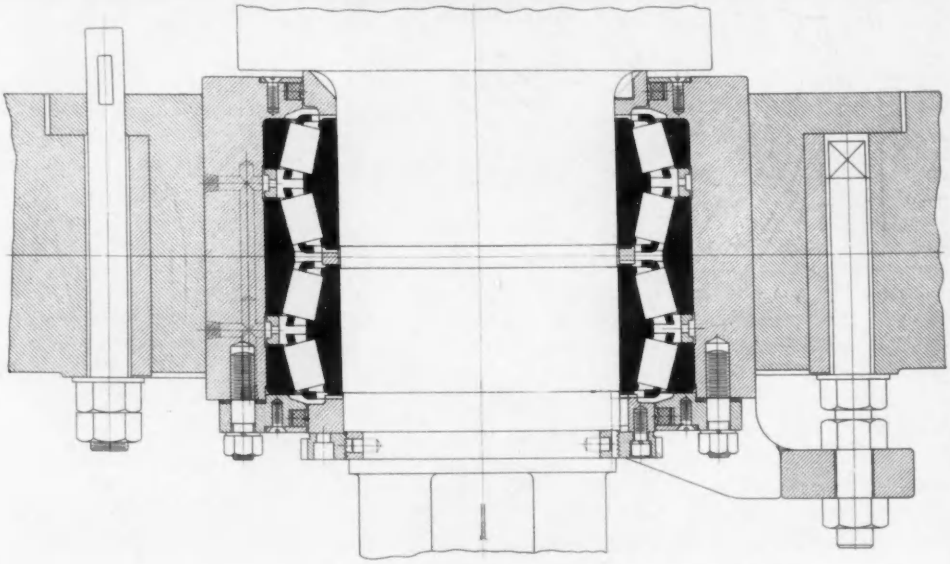
A seamless steel tube piercing mill is adapted for the piercing of billets into long tubular form. The piercing machine has a tremendous capacity for work and is capable of absorbing 2500 hp in the piercing of seamless tubing. In this respect it is tremendously superior to any type of machine-drilling operation as the machine-tool industry provides no means of absorbing this tremendous power input in drilling or machining tubular structures. The typical tube piercing mill consists of two rollers mounted askew to force the billet over the piercing point and the tube mill, while having tremendous production capacity, has certain limitations as regards accurate measurements pertaining to concentricity.

A material improvement as regards accuracy of the work and control of the product is afforded by the three-roller Assel Elongating Mill. This machine holds concentricity within narrow limits, practically machining limits, and provides a product that can be ground directly from the piercing operation. It is now in process of being adapted to the production of shells, the product being of such

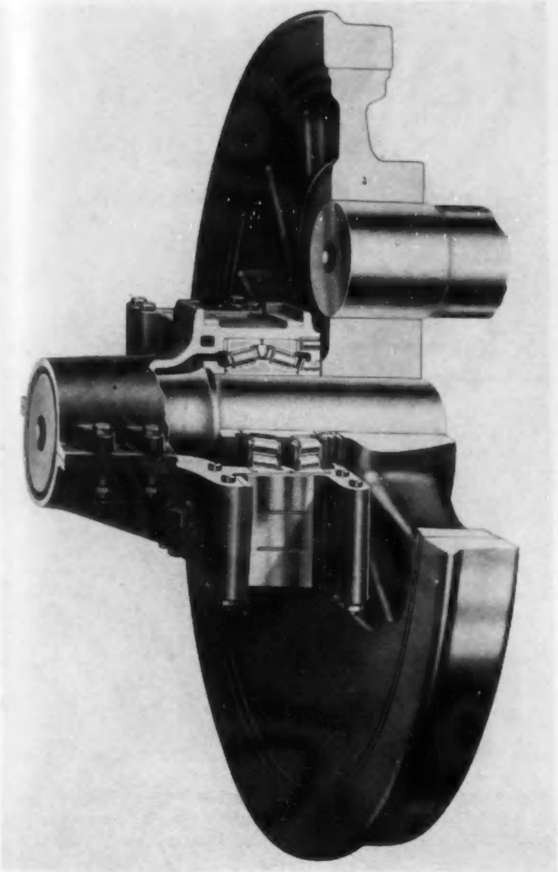
▪ Fig. 14 – End view of roller-bearing four-high steel mill – Tapered roller bearings are used on all roll necks, screw-downs, pinions, reel drives, and coil centers. Back-up roll necks are equipped with four-row tapered roller bearings, 30 x 48 x 32 in.



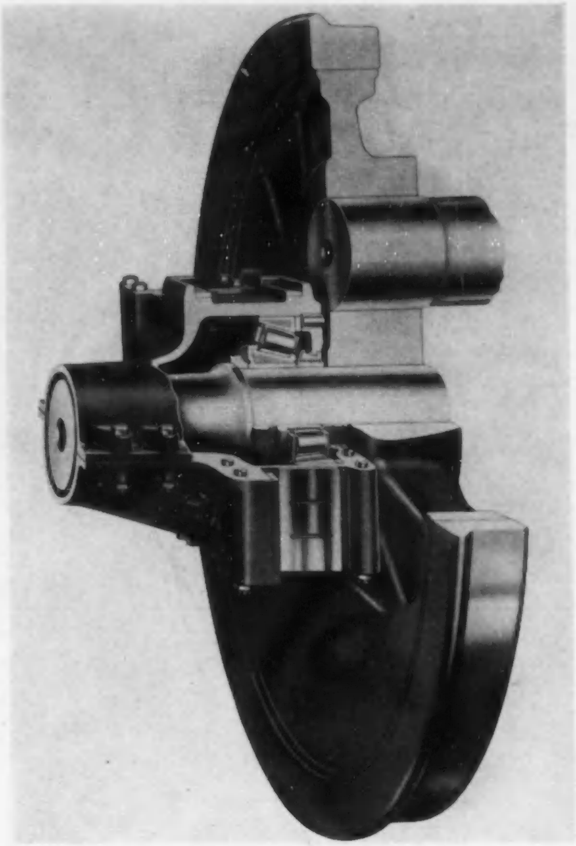
▪ Fig. 15 – Back-up roll for a 96-in. plate mill (four-high) with tapered roller bearings in chocks mounted at both ends ready for installation in the housing – Bearings are 29½ x 46½ x 29 in.



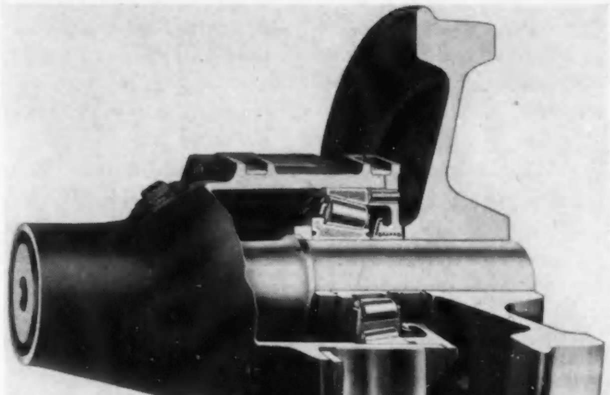
▪ Fig. 16 – Typical roll neck mounting – The left side of centerline shows the floating end of the top or bottom roll. The right side shows the fixed end



■ Fig. 18 – Tapered-roller railway bearing – steam locomotive driving axles – split-type tubular housing application with double bearing for existing and new locomotives



■ Fig. 19 – Tapered-roller railway bearing – steam locomotive driving axles – split-type tubular housing application

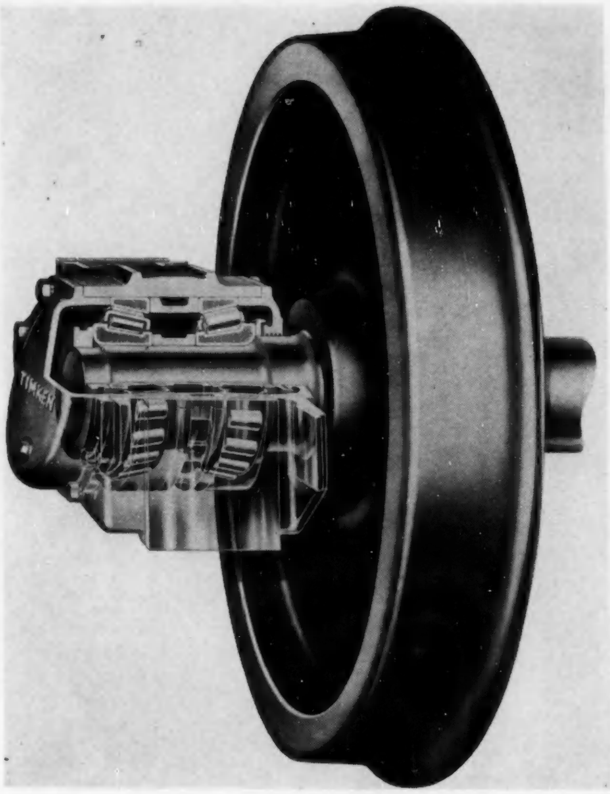


■ Fig. 17 – Tapered-roller railway bearing – steam locomotive engine truck one-piece tubular housing application

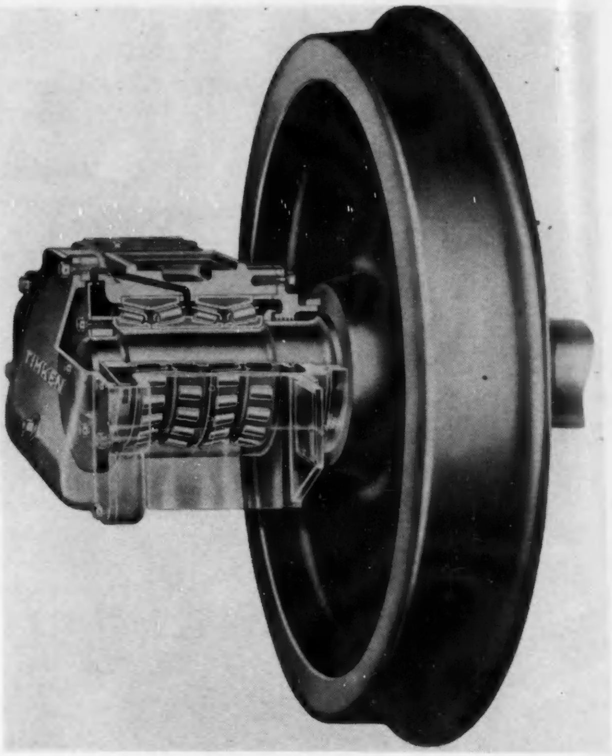
accuracy in the hot piercing operation that a saving of 36% in the weight of the shell billet is effected.

The railway engine truck mount is shown in Fig. 17. This application consisting of an axle, pair of wheels, one-piece housing and two bearings, floats as a unit supporting the front end of the locomotive through the medium of a truck frame and truck springs. An absolute alignment of housing elements, particularly the bearings and axles, is maintained regardless of irregularities of the track, speed of operation, weather conditions, and temperature. An allowance is made in the bearing design for the flexure of the axle under load, the normally loaded axle having 75% full line contact on the rolling elements. This application illustrates the most compact and successful railroad application. It has operated millions of miles with an almost complete absence of bearing, axle, and housing trouble. The life of the wheels is extended materially. Bearing inspection is now limited to replacements of wheels which generally occur every several hundred thousand miles.

In a single bearing driver application the mounting is similar to the engine truck mounting as regards alignment and protection of parts and maintenance of bearing contact with wide variation in load and irregularity of movement of the housing assembly over the track. The housing is split on the horizontal centerline to facilitate assembly, inspection, and servicing. The bearing is reversed with the cone impinging through a spacer against the wheel hub. It is a very successful and economic bearing mounting. The single-bearing application requires a wider pedestal opening than is generally available in the plain bearing



■ Fig. 20 – Dual tapered-roller railroad journal application for locomotive trailers and tenders and passenger-car equipment



■ Fig. 21 – Four-row tapered-roller bearing railroad application which interchanges with all sizes of friction bearings and fits into existing trucks without change

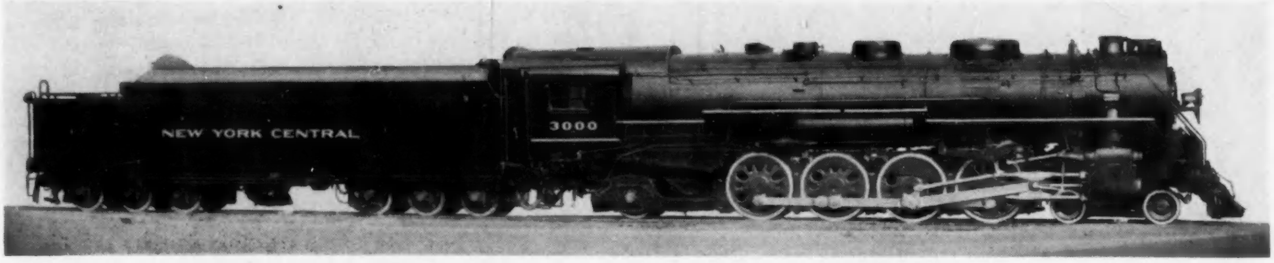
locomotive and is therefore used entirely on new locomotive construction.

The double-bearing locomotive driver application is shown in Fig. 18. Bearings are mounted as a pair for each wheel. The overall dimensions of the bearings are reduced materially. The bearing capacity is equivalent to the single bearing application by using about three times the number of smaller rollers. The space requirements enable this mounting to be used in locomotives originally designed for plain bearings, and more than half of the locomotive driver bearing applications have been made replacing plain bearings, providing the advantages as regards lower maintenance, increased capacity, greater mileage to existing locomotives otherwise suitable for long years of continued work. This application has practically revolutionized steam locomotive construction. Locomotives so provided are enabled to do 50 to 100% more work measured in tonnage and mileage. The double bearing requires the split housing for its economical application.

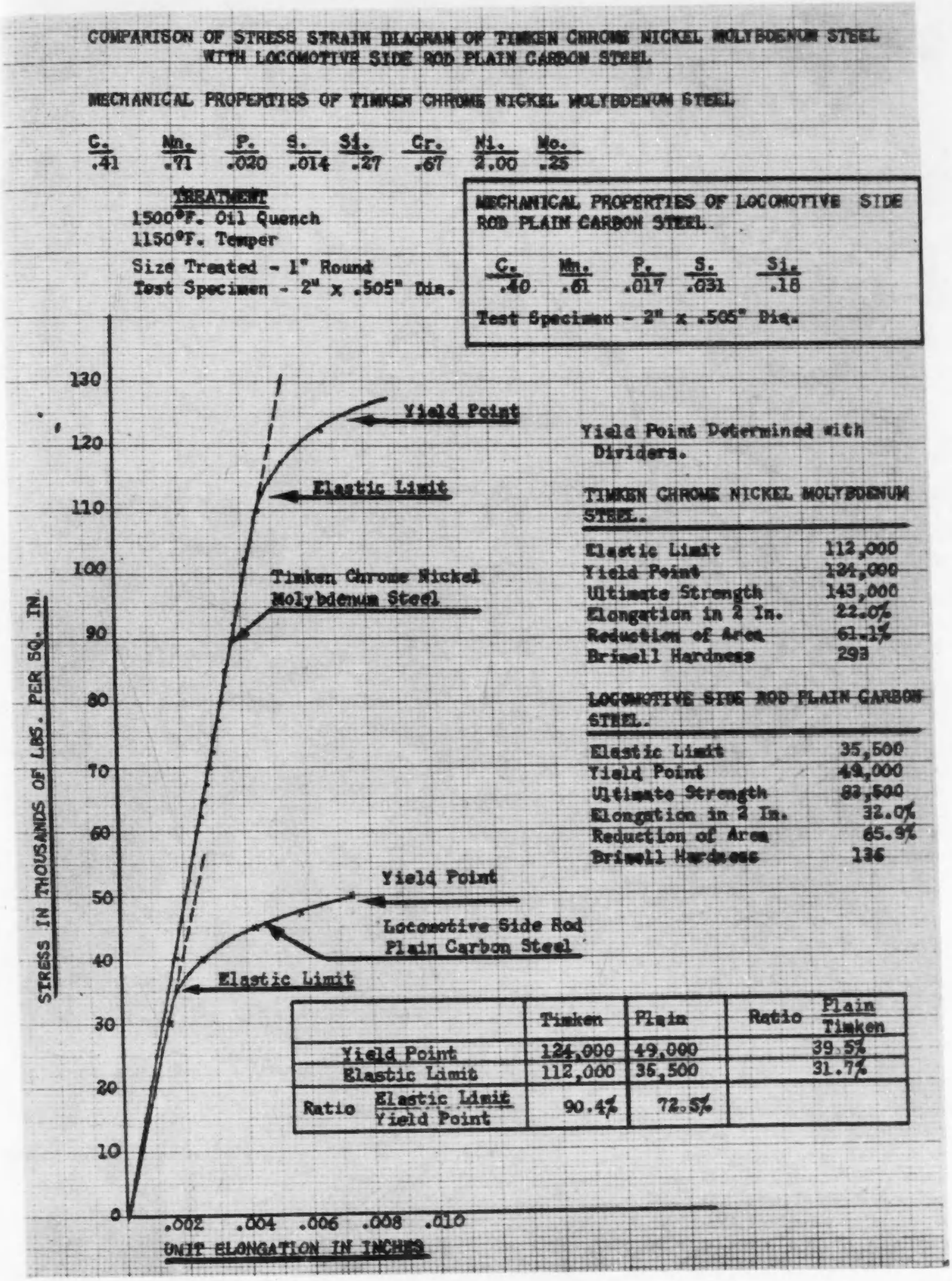
Single-bearing driver construction is illustrated in Fig. 19 and shows split housing with integral head and bearing impinging against shoulder on axle. The construction is favored in many cases by the builder as the wheel is mounted entirely independent of the bearing.

The outboard axle bearing is shown in Fig. 20. The housing is mounted on a pair of bearings and is applicable as a direct replacement on locomotive trailers and tenders and in passenger equipment. It is widely used in these services. It interchanges directly with the plain bearings on locomotive trailers.

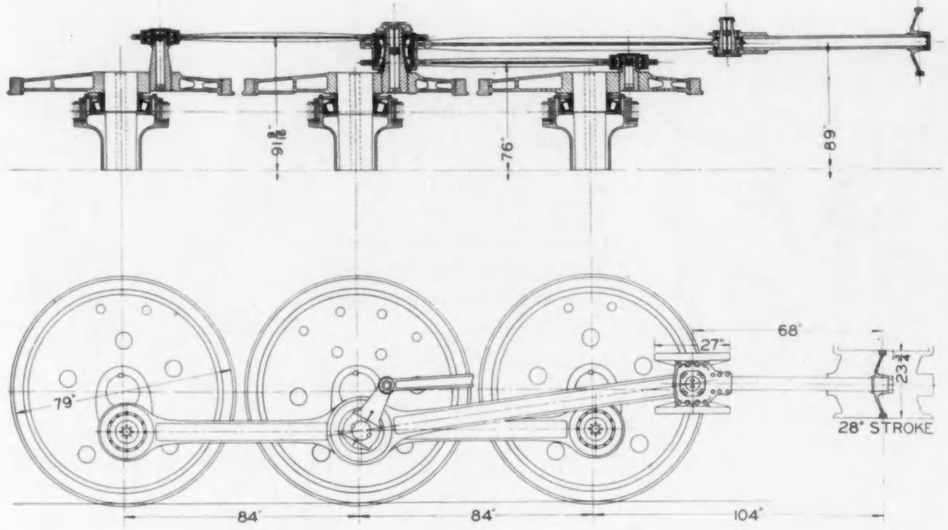
The quad bearing is shown in Fig. 21. This is a most useful application as it interchanges within the narrow pedestal width limitation on locomotive tenders and passenger-car equipment. The relative small size of the



■ Fig. 22 – Combination freight and passenger locomotive



■ Fig. 23 – Comparison of stress-strain diagram of Timken chrome-nickel-molybdenum steel with locomotive side rod plain carbon steel



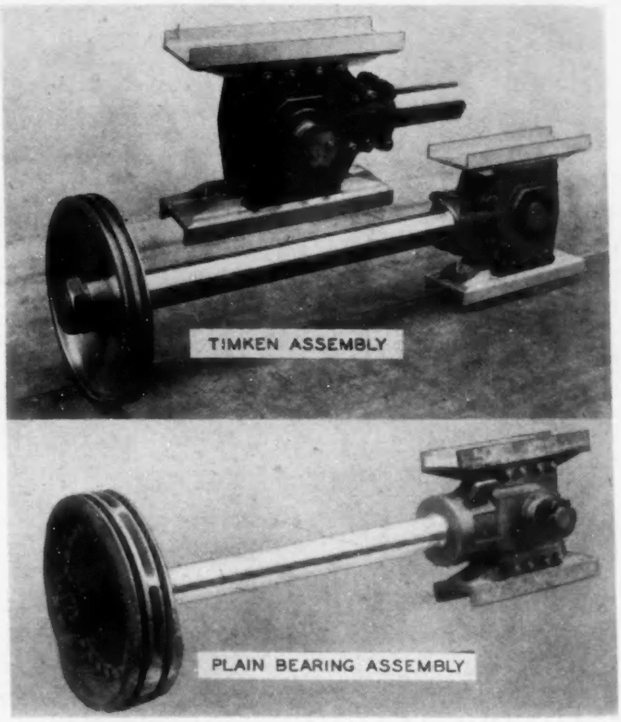
■ **Fig. 24 – Locomotive reciprocating parts – main and side rods, rod bearings, wheels, axles and axle bearings on a 4-6-4 passenger locomotive**

rollers indicates the use of an oil lifter for the distribution of oil over long periods, measured in hundreds of thousands of miles, in bearing life. This and the preceding application are well adapted to heavy military equipment.

The first complete roller-bearing locomotive was built by the company to demonstrate the advantages of roller-bearing equipment in railroad and particularly locomotive operation. It was operated throughout the United States, being loaned without charge to 14 of the principal railroads. It demonstrated an outstanding capacity to handle heavy freight, fast freight and fast passenger service. Nearly all locomotives built today embody the principles originally applied on this locomotive. The reluctance on the part of railroad managements to inaugurate the general use of roller-bearing equipment on a heavy unit, such as locomotives, was eliminated entirely by the loan and operation of this locomotive. The driving-wheel diameter is materially smaller than conventional passenger locomotive practice, but the freedom with which drivers can be rotated at higher relative speeds makes up for this reduced size, and at the same time, provides the increased pulling power necessary for freight traffic. The starting resistance of a roller-bearing locomotive is about one-tenth that of a plain-bearing locomotive and this adds to the starting power, reducing, and generally eliminating, the shock in passenger equipment in getting under way. Years of service have now demonstrated that the normal expectation of reduction in maintenance of roller-bearing locomotive is about 40%, and this is particularly valuable in view of the increased performance in monthly mileage varying from 30 to 130%, a generally expected average in heavy service being 70% more work. It is in keeping with the demands that an increase in performance of 50%, and often more, is obtained at an increased cost of about 10%.

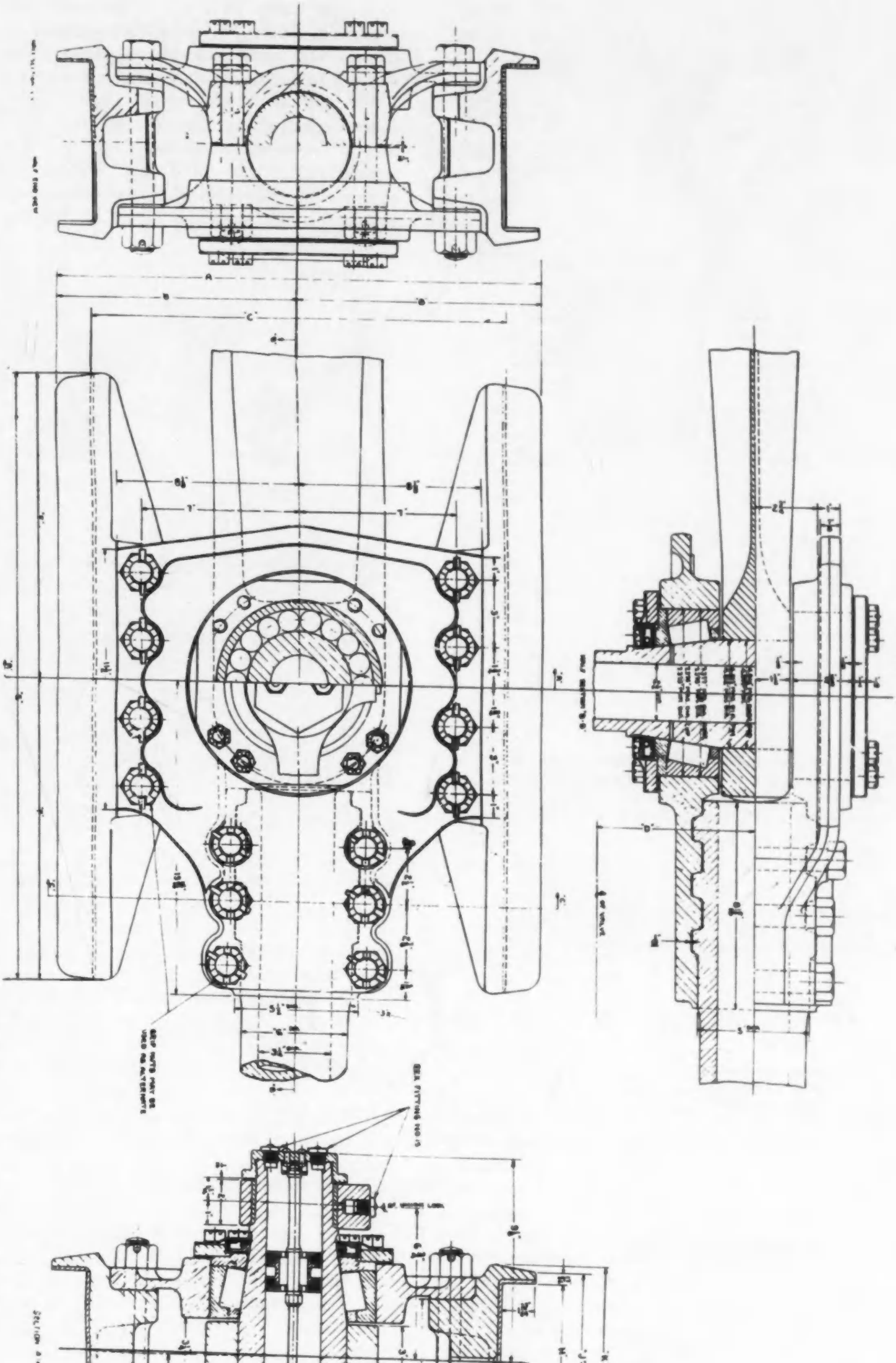
A fleet of 50 New York Central locomotives are completely equipped on all wheels and crossheads, and 5 of this fleet have roller-bearing rods.

A combination freight and passenger locomotive is shown in Fig. 22. This is one of 50 New York Central new Mohawk class, of which 25 are engaged in the heavy-



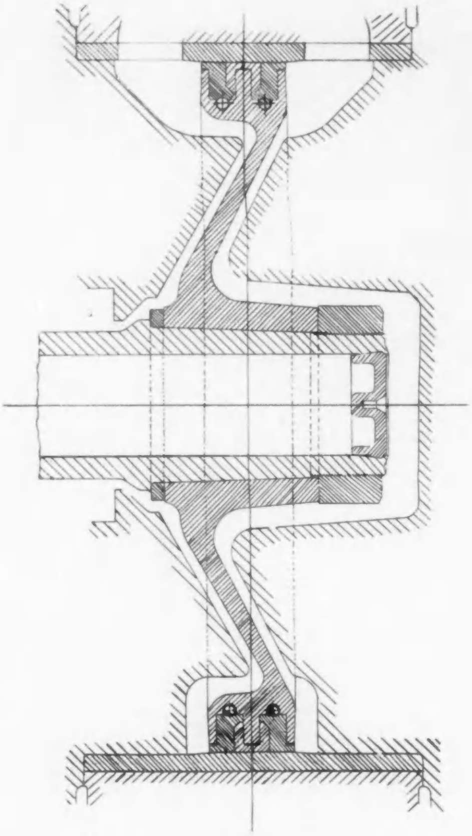
■ **Fig. 25 – Comparison of reciprocating weights, lb**

	Plain bearing	Timken bearing
Crosshead assembly	754	367
Piston, piston rod & parts	765	350
Front end of main rod	422	210
Union link and bushing	30	17
Total	1971	944
Per cent	100	48

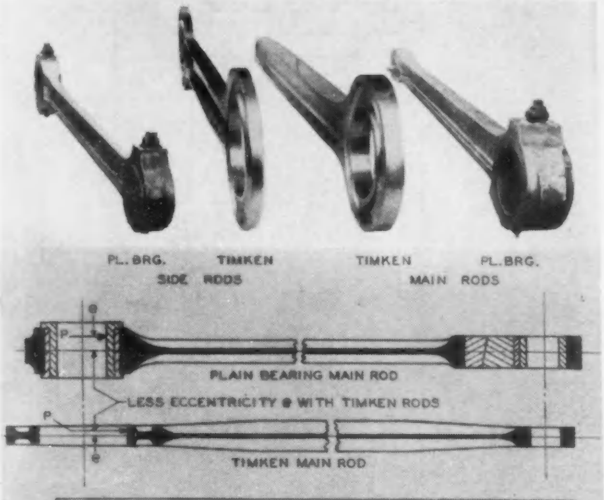


■ Fig. 26—Crosshead assembly showing integral key engagement between two-piece crosshead and tubular piston rod, aluminum-alloy drop-forged slipper with twin rubbing lining. Two roller bearings transmit the load instead of one plain bearing. Each roller bearing is backed up by two oil-retaining bronze packing rings. The small diameter, narrow width, and light weight of main rod front end are shown

and fast-passenger traffic of this system, the balance being in freight traffic. It is one of the most useful locomotives ever built. It can be used interchangeably in either freight or passenger service, and often a group of these locomotives come in to a terminal on passenger service and leave within a few hours in freight service, and to this extent the locomotive effective inventory is increased materially. A locomotive driver-and-wheel balancing system was developed especially for this locomotive following several years' research on this subject. The driving wheels are the standard freight wheel size of 69 in., but the reciprocating

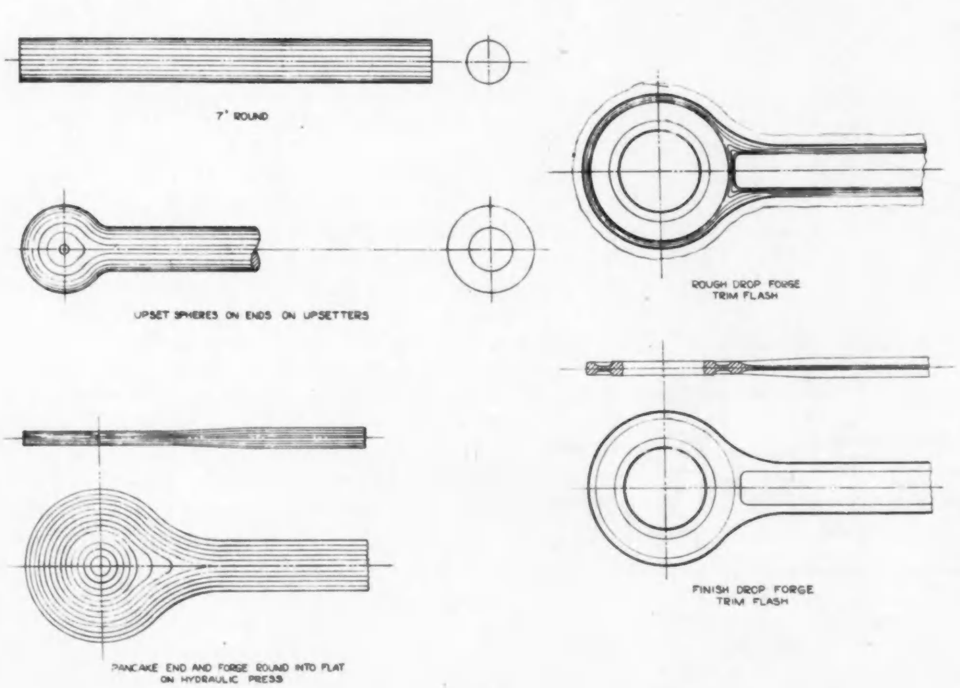


■ Fig. 27 – Section of piston and piston rod end showing light-weight hub section, thin spring-tempered web section, and light sections of rim to receive packing application on hollow piston rod

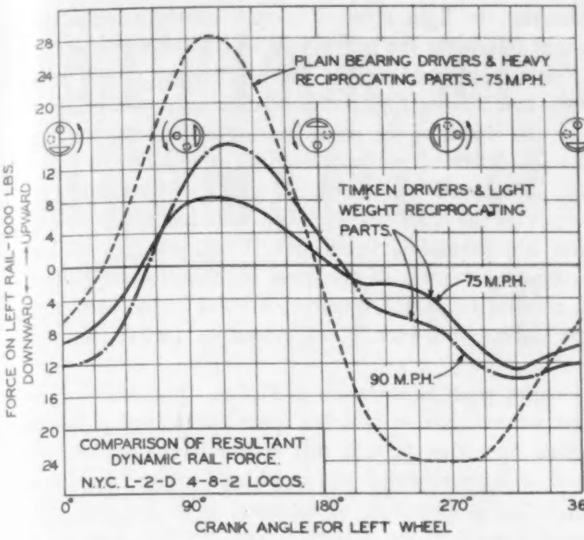


COMPARISON OF ROD WEIGHTS, LBS.				
	SIDE RODS		MAIN ROD	
	PLAIN BRG.	TIMKEN	PLAIN BRG.	TIMKEN
WT. ON MAIN PIN	556	338	581	319
WT. ON FRONT PIN	190	150		
WT. ON REAR PIN	176	150		
WT. ON X-HEAD PIN			422	210
TOTAL	922	638	1003	529

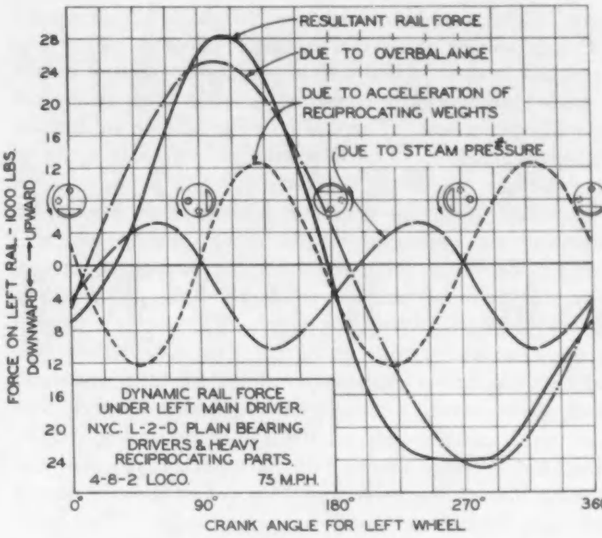
■ Fig. 28 – Rod sections and weight comparison – main and side rods of conventional and new design. The table shows great reduction in weight although strength is greater. New rod ends are approximately one-third the width of the conventional plain rod ends. Column strength is increased markedly due to less eccentricity



■ Fig. 29 – Forging practice – development of rod forging from round bar upset to spherical, pancaked to round flat end, then drop-forged to H-section eye. Note the uniform grain flow around ends continuous with flange section



■ Fig. 30 – Comparison of resultant dynamic rail force

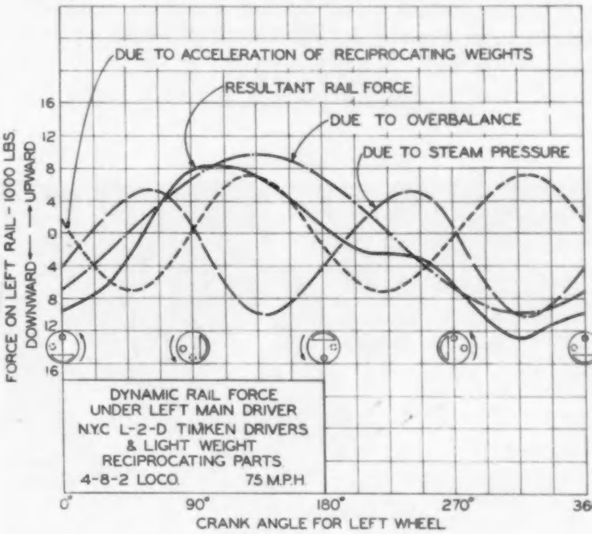


■ Fig. 31 – Dynamic rail force components under left main driver

parts, comprising piston, piston rod, crosshead and front end of main rod, are reduced in weight nearly one-half. The conventional practice has been to balance reciprocating parts 35% to 50% but, having reduced the weight approximately this much, an entirely new balancing system is available. This outstanding reduction is utilized to reduce the vertical rail blow, dynamic augment, about two-thirds, applying the balance to reducing the nosing – the tendency to snake along the track. The vertical reactions of the drivers are therefore only one-third as great, and the nosing about one-half as much. The final outcome is one of the ablest and smoothest-operating locomotives in the 80-mph zone that the American railroads have thus far developed. The development of this locomotive will go a long way in keeping the steam locomotive in the transportation picture during the life of the present generation.

A 4-cyl locomotive is the outstanding contribution of the Pennsylvania. It uses four engine sets on the 8-coupled drivers, there being four sets of pistons, piston rods, crossheads and main rods with a corresponding reduction in piston thrust and cylinder size. The experience with this type of locomotive will be watched with much interest by the railroad industry, and it has much of promise for high-speed locomotive operation.

Comparison of high dynamic and plain carbon steel is shown in Fig. 23. A medium alloy steel using the best properties of molybdenum, chromium, and nickel with a medium carbon is illustrated comparatively with plain carbon steel. The useful property of a steel is the yield point, and for the same weight this is increased from 55,000 to 124,000 psi. The deflection within the yield point is identical with carbon steel per unit of stress, but the yield is practically 7 units as against 2½ on the carbon steel within the safe working zone. This outstandingly valuable steel, developed primarily for railroad and heavy automotive equipment, has been adopted in its entirety by the aviation industry and has been found the most reliable strong steel with high dynamic properties for the heavily stressed members in military and commercial airplane devices, comprising crankshafts, connecting rods, propeller shafts, propeller hubs and a multiplicity of small parts. The demands of



■ Fig. 32 – Dynamic rail force on new locomotive to same scale as Fig. 31

this steel have increased over 400% with the advent of our defense program.

Locomotive reciprocating parts are shown in Fig. 24. This figure illustrates a sectional view of a six-coupled locomotive showing the driver bearings, main and side-rod bearings, crosshead bearings and reciprocating parts, comprising the piston, piston rod, crossehad and front end of the main rod. The reciprocating parts have been widely adopted. The crankpin bearings are being used in a limited way and have shown outstanding capabilities as regards higher permissible speeds. Locomotives with this equipment have been operated on test as high as 140 mph without damaging the rail. It is an application that will be followed with much interest but, being a tremendous development, it will require considerable time to perfect.

Reciprocating weight comparison is shown in Fig. 25, the lower view being the plain bearing and the upper the

light-weight assembly. The new weight is 944 lb against a former weight of 1971, a reduction of 52%. This is of particular interest in that 1 lb saved in reciprocating weight at high passenger-train speeds reduces the blow on the rail by 72 lb per wheel. There is no other known investment that pays 7200%. The development bids fair again to revolutionize the steam locomotive.

Crosshead assembly is shown in section in Fig. 26, illustrating the crosshead forged in two halves which is bolted to the piston rod through the medium of the integral keys. There is no variation in length of the piston-rod assembly through take-up for wear. The steam clearance in the cylinder remains uniform. The construction provides for the use of drop-forged aluminum-alloy crosshead shoes, these having a weight one-third that of steel, thus effecting a material reduction in the reciprocating weight. Crosshead bearings of the full-roll type are mounted one on each side of the main rod and operate directly on the inner race which constitutes the crosshead pin. This crosshead bearing is relatively small in size and yet carries rapidly alternate loads of the magnitude of 120,000 to 160,000 lb, alternating at high speed of 12 to 14 per sec. It demonstrates in an effective way the tremendous capacity of a relatively small roller bearing to absorb punishment and illustrates a mounting that should be found invaluable in military service.

The piston-rod sectional view is shown in Fig. 27, and illustrates the light sections and outstanding reduction in weight following the proper use of high dynamic steel.

Rod sections and weight comparison are shown in Fig. 28. The roller-bearing rod is relatively narrow, being one-third the width of the plain bearing rod on the main pin. The possibility of eccentric loading is reduced to that extent. The main rod weight is 529 lb as compared with 1003 lb on the plain bearing rod. Side rod weight is somewhat less favorable, being 638 lb against 922 lb. The outstanding feature of the new rods is the complete absence of oil holes, there being no holes in the rods of any kind except the ones at each end to receive the roller bearing.

Forging practice is shown in Fig. 29. The rods are made from a round bar on which a sphere is formed at the end; a press operation flattens this sphere to a pancake; from which it is drop-forged and machined to the finished rod. The grain is therefore continuous along the top flange, around the eye to the bottom flange. Failure through end grain is eliminated.

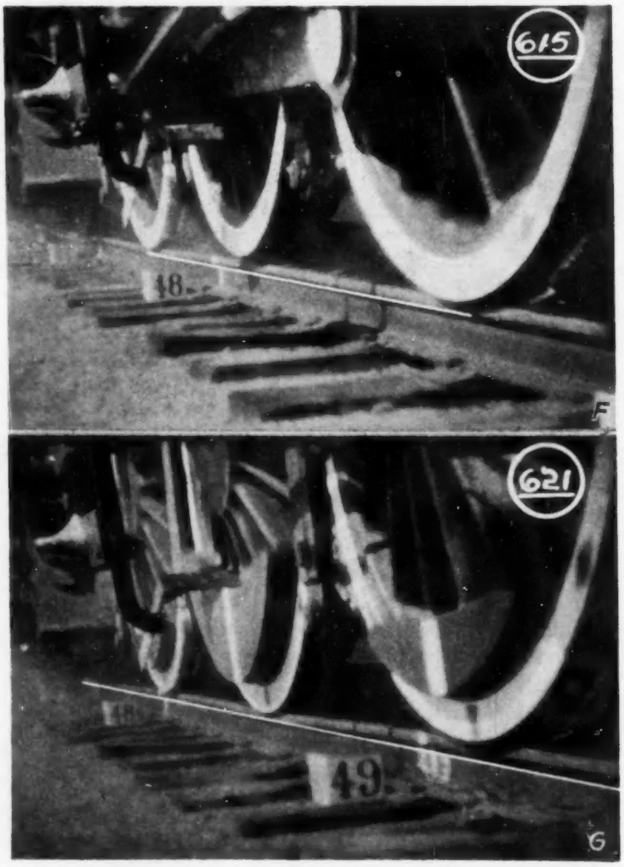
When this rod is applied on the New York Central high-speed Hudson locomotives the position of the main rod is interposed between the two side rods, whereby the load on the main pin is reduced more than one half.

A dynamic rail force comparison is shown in Fig. 30, the upward blow against the springs is reduced from 28,000 to 8000 lb at 75 mph, and the downward blow against the rail is reduced from 24,000 to 12,000 lb. This reduction follows the scientific application of high dynamic steel and light-weight reciprocating parts.

Dynamic rail force components are shown in Fig. 31. This results from the original design on the plain bearing locomotive. Dynamic rail force on the new locomotive to same scale is shown in Fig. 32. This follows the scientific application of light-weight reciprocating parts and modern balancing.

Flexure of rail is shown in Fig. 33. The upper view shows the rail flexure with dynamic augment upward against the spring rigging and shows rail flexure about ½ in. up between No. 1 and No. 3 drivers following the reduced loading on main driver. The lower view shows deflection of the rail downward about ½ in. following the heavy dynamic augment of plain-bearing engine at 108 mph with 315 lb overbalance on the main, the middle wheel. Corresponding forces and rail reactions and rail flexure are reduced two-thirds to 75% by the light-weight reciprocating parts and scientific balancing described herein.

War is the master human activity that embraces all other enterprises and activities of man, and it follows as a natural corollary that the improvement of any division of man's work, such as steel making or machine-tool manufacture, makes a corresponding improvement in our national defense. The roller bearing has improved industrial production to an almost unbelievable extent and has provided a capacity for producing the implements of war that can not be matched by any other nation. These revolutions in industry, comprising the steel mill, the machine tool, material handling, the automotive, the motor truck, the mining industry and the oil-well industry have been made primarily for peaceful use, but American ingenuity and enterprise will rapidly convert these essentials of production to military use and will provide a defensive and offensive military strength unmatchable anywhere in the world.



■ Fig. 33 – Flexure of rail – three drivers, Locomotive No. 3012. (Enlargements from 16-mm films taken at 100 frames per sec showing curvature of rail under main wheel at a slipping speed of 108 mph)

ROLLING RESISTANCE of Pneumatic Tires as a Factor in CAR ECONOMY

by

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THE pneumatic tires consume appreciable though not large percentages of the total power involved in car operation. Numerous factors affect the amount of power so consumed and converted into heat within the material of the tire itself, as follows:

It is increased by an increase of speed, of load, of rim diameter, of traction, and of slip-angle; and decreased by an increase of inflation pressure and of rim width.

Six-ply tires have higher rolling resistance than have four-ply tires. On gravel roads, the power consumption averages more than twice its value on hard road surfaces.

WHEN the chassis is examined from the standpoint of operating economy, it immediately becomes evident that the tires are the predominant if not indeed the only chassis parts which consume appreciable amounts of power. It is the purpose of this paper to present the available data relating to the power consumption of tires, and to portray how it is affected by various operating conditions and by the several trends in automotive design.

It was felt that a consolidation of the data which have been accumulated by various tire companies would furnish a basis for the most comprehensive and authoritative treatment of this subject. To this end, a committee of tire engineers from several companies have pooled and averaged their test results and have collaborated in preparing the following discussion.

In this paper the term power consumption will be stated in horsepower, at the indicated speed. The term tire drag will be in pounds at the indicated speed and tire load. Rolling resistance has been used by various writers to express the drag in pounds *per 1000 lb of load* on the tire; such usage will be followed in this paper.

Most of the data now to be presented were obtained in the laboratories of the contributing companies by running the tires against pulleys or flywheels. Various methods were used to evaluate the power consumption or drag of the tires. Under similar conditions of load, inflation, and speed, the absolute power consumption is somewhat larger on the curved surface of a flywheel than on a smooth, hard road, but *relationships or ratings* are very closely the same.

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 4, 1941.]

Effect of Speed

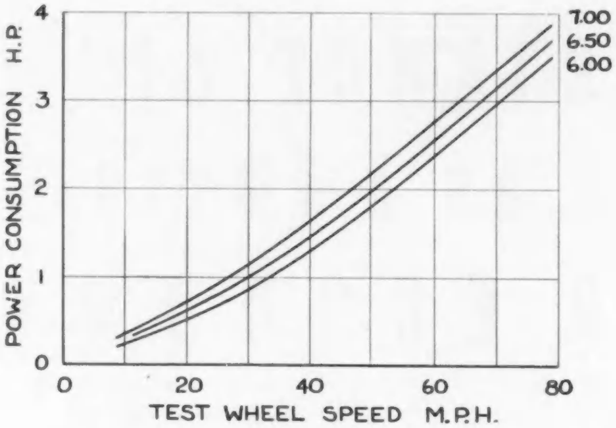
Fig. 1 shows the effect of speed on power consumption. A curve is shown for each of three tire sizes, 6.00-16, 6.50-16, and 7.00-16, with each at its T & R recommended load and inflation, and mounted on its recommended rim. The inflation was maintained constant at the stated value. Before each observation was made, the tire was run long enough for the temperature to "level off."

In Fig. 2 the same data are plotted in terms of tire drag, expressed in pounds. If rolling resistance per 1000 lb of load is calculated, Fig. 3 shows that it is approximately constant for the several sizes.

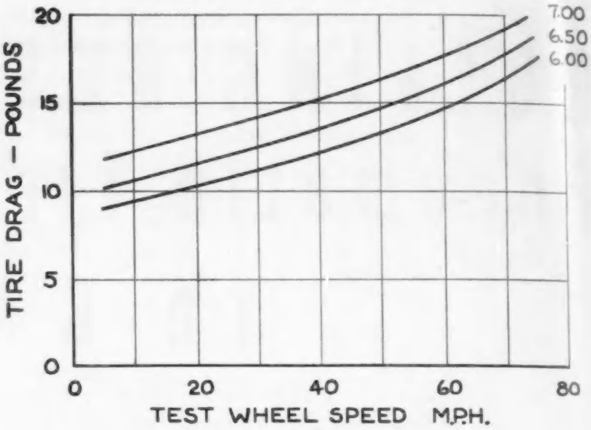
If the pressure were allowed to build up corresponding to the higher temperature at higher speed, the curves of Fig. 2 would be more nearly horizontal, because the deflection of the tire would be less at the increased pressure.

It may be of interest to see how much power is consumed by the tires in several typical cases of vehicle operation.

First, a small car equipped with 6.00-16 tires might weigh 3500 lb with its load. At 30 mph the tires would consume 3.4 hp, and 9.0 hp at 60 mph. A medium-sized car, weighing 4000 lb with load, has 6.50-16 tires. Their consumption is 3.8 hp at 30 mph, 9.7 hp at 60 mph. For a large car, with a gross weight of 4500 lb on 7.00-16 tires, the values are 4.5 hp at 30 mph, and 10.8 hp at 60 mph. All these values are on the high side, for reasons previously explained.



■ Fig. 1 – Effect of speed on power consumption



■ Fig. 2 – Effect of speed on tire drag

■ Rim Diameter

With all passenger-car equipment in 1941 having either 15-in. or 16-in. rims, interest in the effect of rim diameter is limited to the relationship between these two. For a range of speeds and of tire sizes, the power consumption of 15-in. rim tires is from 90 to 95% of its value for 16-in. rim tires.

■ Rim Width

During the past year there has been much interest in exploring the several aspects of the use of wider rims.

TIRE SIZE	AT 20 MPH	AT 40 MPH	AT 60 MPH
6.00-16	11.1	13.2	16.1
6.50-16	10.9	12.9	15.4
7.00-16	11.5	13.3	15.4
AVE	11.2	13.1	15.6

■ Fig. 3 – Rolling resistance per 1000-lb load

Since widening the rim decreases the deflection of a tire, it also decreases the drag or the rolling resistance. On the widest rims currently recommended, the drag ranges from 85 to 95% of its value on the narrowest approved rims, all at the same inflation pressure. This range depends on differences in tread flatness and other features of construction as among different makes and types of tires.

■ Six-Ply Versus Four-Ply Tires

When, because of road conditions or for other reasons, the four-ply original equipment tires are replaced by six-ply tires of the same size, the power consumption usually is increased. Thus, for the same load and same inflation, six-ply tires at 30 mph average 7% more rolling resistance than corresponding four-ply tires. At 60 mph, the differential averages 4%.

■ Effect of Inflation and Load

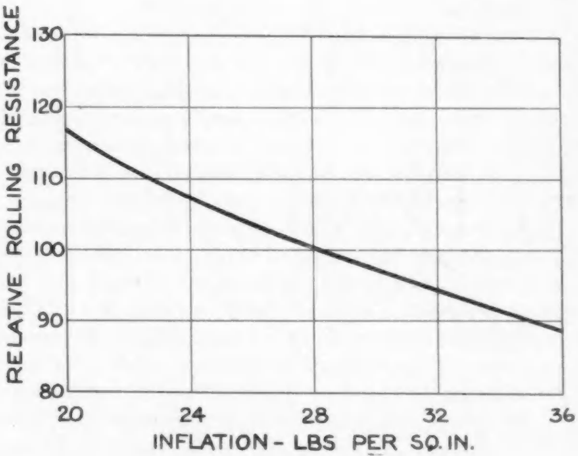
Since rolling resistance is related closely to tire deflection, whatever increases or decreases deflection may be expected to have a corresponding effect on rolling resistance. In Fig. 4 is shown a composite curve, averaging the effect of inflation on a variety of tires and at several speeds. The rolling resistance is arbitrarily rated at 100% for the recommended inflation of 28 psi.

Changing the load has an analogous effect. A 20% increase of load increases the drag or power consumption by 20 to 25%.

■ Rayon Versus Cotton

Rayon has had a certain acceptance in passenger-car tires, as giving a flexible, easy-rolling tire. Fig. 5 presents several comparisons between standard 4-ply cotton tires and similar sized 4-ply rayon tires. The same four tires of each kind were used first for laboratory test; then they were mounted on a car and rated for rolling resistance by coasting tests on smooth, level concrete; finally gas consumption was determined with the car successively running on each set of tires. All observations were at 30 mph.

These comparisons were obtained with steady non-stop



■ Fig. 4 – Effect of inflation on rolling resistance

driving on a straight level road. They would not necessarily apply to other conditions or to higher speeds.

New Versus Worn Tires

When tires become worn down, the deflection is reduced, and less material is subject to flexure. This change is reflected in a lessened rolling resistance. With the tread design about two-thirds worn away, the rolling resistance averages 84% of its value for the new tire.

Effect of Road Surface

Power consumption of tires appears to be affected to only a very slight, if not indeed negligible, degree by the differences of texture of the usual hard road surfaces. When such roads are wet, the power consumption is slightly greater, partly because the tires run somewhat cooler.

Prof. Moyer, of Iowa State College Experiment Station, has made available to us some results of his extensive road tests on various kinds of highway surfaces. He reports an average rolling resistance on hard road of about 8 lb per 1000-lb load. The rolling resistance on gravel ranges from 12 to 60 lb, the lower value for dry, firm, well-packed gravel, and the higher for wet, loose gravel.

Prof. Moyer also reports that the all-season average of gasoline consumption, over a wide range of speeds, is approximately 2 mpg greater on gravel roads than on hard roads, and that this corresponds to an average "all-season" rolling resistance on gravel roads of approximately 18 lb per 1000 lb, as compared with 8 lb for hard roads.

Effect of Traction

When a tire is not only carrying its load but delivering tractive effort as well, the rolling resistance is somewhat greater than when it is free-rolling. Fig. 6 indicates the magnitude of this effect when the speed is maintained constant. This graph is the average for the same three tire sizes as in Figs. 1, 2, and 3, with each at its recommended load, inflation, and rim width.

[1] See SAE Transactions, August, 1939, pp. 344-350: "Tire Behavior in Steering," by A. W. Bull.

	COTTON	RAYON
AVERAGE DRAG-4 TIRES BY LABORATORY TEST	100 %	87.0 %
ROLLING RESISTANCE OF CAR BY COASTING TEST	100	95.8
GASOLINE CONSUMPTION	100	96.5

Fig. 5 – Rolling resistance of rayon versus cotton

Effect of Slip-Angle

The increased demand for power in taking a curve is well known. This result is largely because the rolling resistance of the tires is increased by slip-angle. In May, 1939, Dr. A. W. Bull presented a paper before the Society, "Tire Behavior in Steering,"[1] in which he showed, among other things, the effect of slip-angle on power consumption. Fig. 7 is a reproduction of Dr. Bull's chart portraying this relationship.

It may be of interest to see what this factor means in terms of the car itself. The Chrysler Corp. has made available the following results on torque requirements in driving a car around a circle: The size of the circle was such that the full power of the engine was required to maintain a speed of approximately 22 mph, with correspondingly large slip-angles, attitude change, and roll of the car.

At 10 mph, for which only very low slip-angles are developed, the torque requirement only slightly exceeds that for straight-ahead travel at the same speed. At 15 mph, the circle required approximately 50% more torque than straight-ahead. At 20 mph, which approached the limiting speed at which the car could be held in the circle, the torque required was more than three times as great as was needed for straight-ahead driving at the same speed.

These results are quite closely in harmony with the comparisons which could be derived from Fig. 7.

In submitting these results and comparisons indicative

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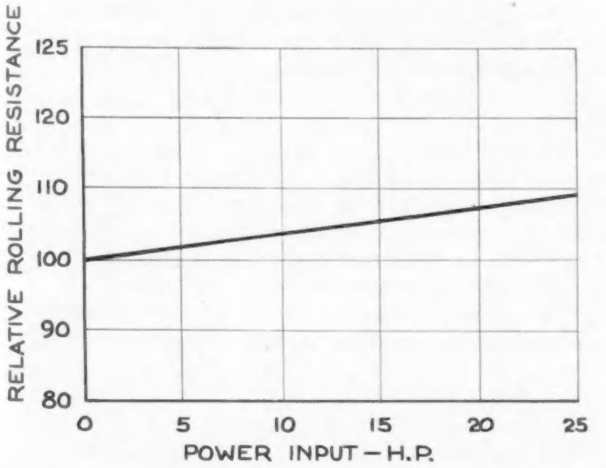


Fig. 6 – Effect of traction on rolling resistance

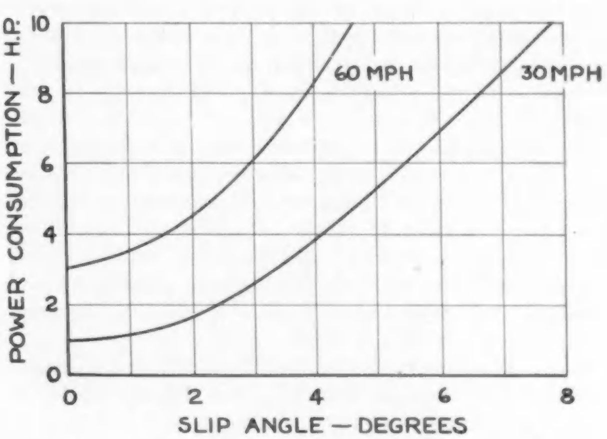


Fig. 7 – Effect of slip-angle on power consumption

ICING PROBLEMS in A

HEAT and alcohol are the most common de-icing mediums, but they offer the disadvantages of loss in power and increased weight. The protected air intake is effective if the rate of water ingestion can be reduced during precipitation and the air temperature raised high enough to be effective during severe icing conditions.

The ideal induction system is one which requires none of these, and work is being done in hopes of reaching this goal.

The research program was initiated by the Special Subcommittee on Induction System De-Icing of the NACA. The project was set up at the Bureau of Standards, as they were well equipped to handle this sort of work. The apparatus is described and the method outlined for establishing the icing conditions.

Types of induction ice are described and pictures of formations found during tests are shown. Preliminary results are summarized as follows:

1. The rate of ice accretion was roughly proportional to the rate of water ingestion.
2. Through the lower temperature range, the smaller the droplet size, the faster the rate of ice accretion for a given rate of ingestion.
3. The most dangerous condition was found to be around 30 F air temperature and small droplet size.

Design criteria are set up as a result of this preliminary work which are the basis of rates for making a system which would be substantially ice-free.

Some new developments are described, such as two types of flush bulb thermometer and two icing indicators which would give the delayed action favored by the industry.

■ ■ ■

THE history of carburetor icing dates back over 20 years. The earliest report which we have is a paper published in 1920 by S. W. Sparrow, then of the National Bureau of Standards, in which he described the phenomenon and pointed out that many unexplained airplane crashes almost certainly could be attributed to this cause. He described the icing action as giving the pilot the impression that: "A demon was operating the throttle."

During the ensuing years much has been written on the subject and many solutions offered. Aircraft are now able to navigate in very bad weather, and it is vital that their operation should not be limited by a very insignificant but aggravating matter of induction-system icing.

The problems brought about by the icing phenomenon have always seemed easy of solution, but the often unaccounted for factors involved have prevented a complete solution of these problems.

Heat is the most commonly used ice-preventing means, but this method involves a loss in power, and full-cold operation has been found preferable.

Alcohol is used when the ice has once formed, but this means involves extra cost for equipment and servicing, and extra weight to be carried around.

The alternate protected air intake is effective if the rate of water ingestion can be reduced and the temperature raised high enough to be effective during severe icing conditions.

The ideal induction system for which we strive is one in which no ice will accumulate during severe icing conditions and which obviously would require no heat, alcohol, or protected air intakes during these conditions. This result seems remote in the light of past experiences; however, I feel that progress toward a solution is being made.

Before going on with the details of the problem, it is in order to give a little of the history of the Induction System Icing Project at the National Bureau of Standards and a description of the apparatus used in the present investigation.

During a discussion of induction-system icing problems taken up at the Engineering Maintenance Conference of the Air Transport Association in July, 1940, it was decided that a research program should be initiated to investigate the problem. A committee was formed to act under the Power Plants Committee of the National Advisory Committee for Aeronautics to be known as the Subcommittee on Induction System De-Icing.

Funds for the investigation were provided by the Army, Navy, Civil Aeronautics Authority, and National Advisory Committee for Aeronautics.

At the suggestion of Dr. H. C. Dickinson, the large Altitude Laboratory at the National Bureau of Standards was chosen to house the project as it offered the best available facilities to carry on the investigations.

This altitude chamber is well fitted for the project as it is equipped to produce all sorts of atmospheric conditions. It has an ample air supply, sufficient to test the induction systems of the largest aircraft engines built.

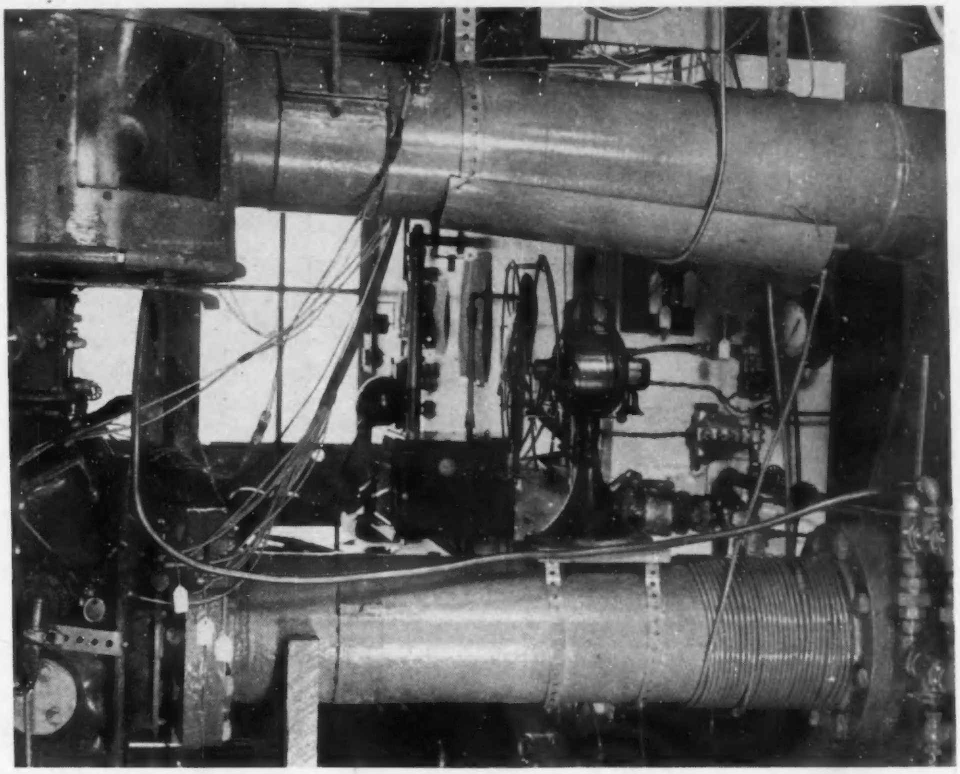
The air supply is carried through the carburetor by suction and is cooled by first conducting it through the refrigerated rooms above the chamber, each having several tons of coils therein. Refrigeration is supplied by two 25-ton compressors and a booster, bringing the total capac-

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Oct. 30, 1941.]

Aircraft Induction Systems

by LEO B. KIMBALL

Senior Engineer, National Bureau of Standards



■ Fig. 1 – Complete set-up of test apparatus

ity to 100 tons. Two additional machines of 100 tons each are being installed to increase this capacity.

The altitude chamber may be sealed and the pressure reduced to correspond to any desired altitude. All observations and control can be accomplished from outside the chamber.

■ Research Program

The research program as set up by the NACA subcommittee includes the investigation of the various variables which affect carburetor ice. These can be enumerated as follows: air temperature, water content of air, droplet size, mixture ratio, air flow, pressure drop through carburetor, metal temperatures, throttle opening, and altitude.

Recording instruments are used to take most of the readings as it is impossible to make observations of all of the variables fast enough to obtain accurate results.

■ Apparatus

A brief description of the apparatus used in the investigation is as follows:

Gasoline flow is measured by the use of rotameters which have a range up to 1600 lb per hr. A record is also made on a continuously recording differential pressure gage.

Metal temperatures are recorded on two 12-point recording potentiometers which give a complete record of the temperatures in 45 sec.

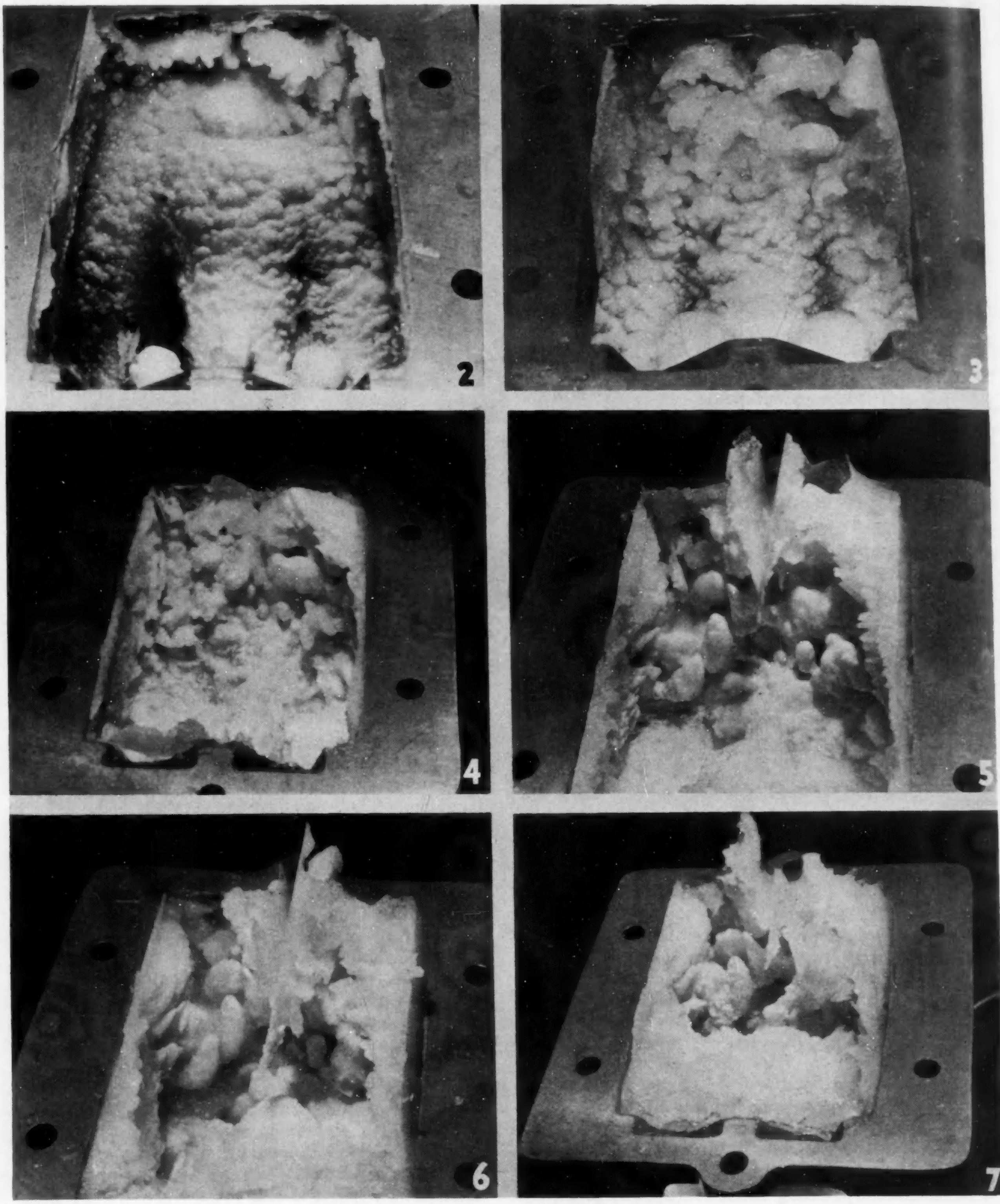
Air flow is measured on interchangeable flat-plate orifices and recorded on another differential pressure gage.

The relative humidity of the incoming cooled air is measured by means of wet-and-dry-bulb thermometers. A sample is drawn through a tube into a Tee, one side leading to the wet-bulb and the other to the dry-bulb thermometer. The tube is heated slightly so that any excess of water in small particles will be absorbed into the air and the actual water content can be ascertained. The correct amount of water can then be added to agree with the condition requirements. While runs are being made at lower temperatures, the cooling coils remove a varying amount of moisture from the air, depending on the thickness of the frost accumulation. It is necessary to make frequent determinations of the moisture content so that the total water content will not change appreciably.

To simulate raining conditions, a nozzle bar is placed in the air duct ahead of the scoop entrance. This bar is connected to a water supply line through a rotameter so that the flow can be controlled accurately. A grid is installed ahead of the nozzle bar to discourage turbulence and reduce wall flow induced by impingement of the water on the sides of the pipe. Droplet size may be controlled by means of interchangeable jets screwed into the nozzle bar. The water is cooled before it enters the spray nozzles to keep it near the temperature of the incoming air. A side view of the complete set-up is shown in Fig. 1.

■ Types of Induction Ice

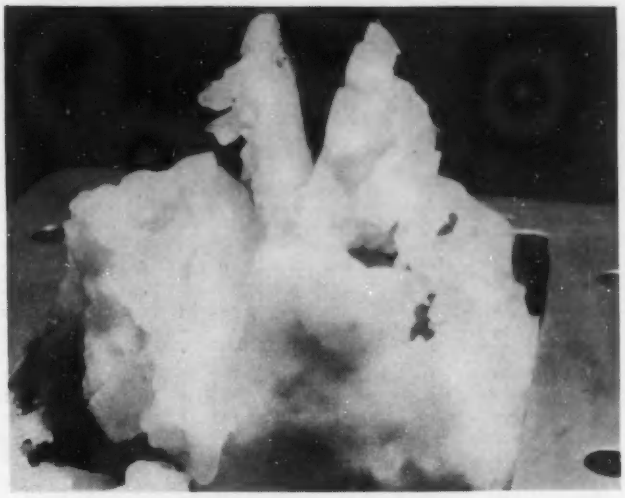
Two distinct types of ice formation are encountered in aircraft: first, a type known as impact ice, which occurs along the entering edges of all exposed surfaces and around the carburetor intake when temperatures are near the freezing point. This formation results from the fixation of semi-plastic ice particles or super-cooled droplets which



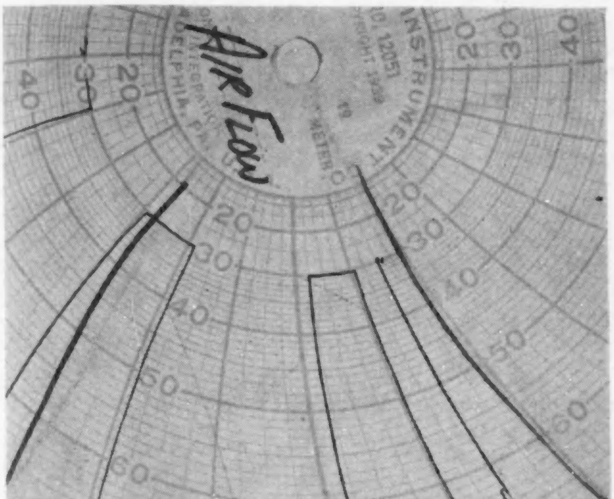
■ Figs. 2-7 – Progressive series of ice formation taken under normal icing at cruising conditions (Pictures were taken during a total of 10 min operation by taking the carburetor off at certain time intervals)

freeze when they impinge against cooled surfaces. There are exceptional conditions in which impact icing has been reported as low as -44 F, indicating that water droplets may exist in a greatly super-cooled state without freezing at this low temperature. This type of icing occurs on the upstream side of the throttle and is quite dangerous as it forms very quickly and during conditions when induction icing would be least expected. Fuel-injection and diesel engines are subject to this type of icing.

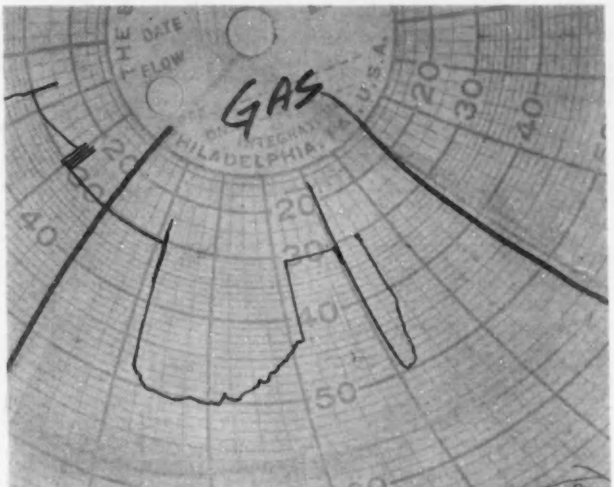
The second type of icing is that with which this investigation is mainly concerned and is known as refrigeration icing. This type of formation occurs mostly beyond the point in the induction system where the gasoline is mixed with intake air. The icing here is due to the refrigerating effect caused by the latent heat of vaporization of gasoline. Since this may result in a drop in temperature of from 40 to 70 F or more, it would cause the precipitation and freezing of practically all of the moisture in the air.



■ Fig. 8 – Complete ice formation taken under normal icing at cruising conditions (same conditions as Figs. 2-7)



■ Fig. 9 – Graphical record of air flow taken during two runs at 5 and 10 g per cu m rain density

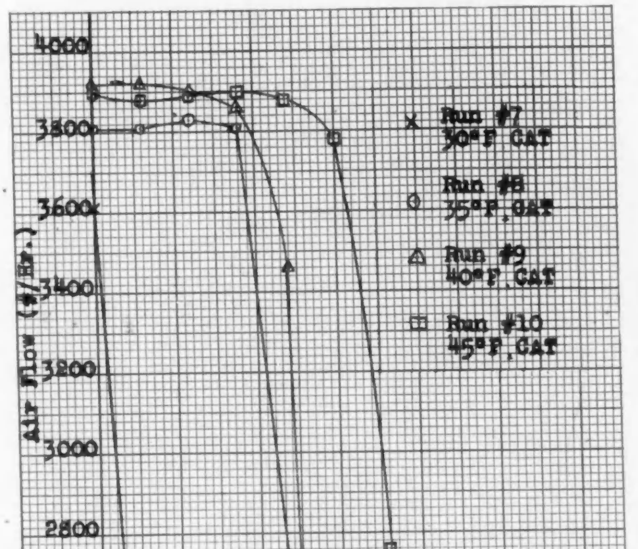


■ Fig. 10 – Gasoline flow chart for same runs as air-flow chart, Fig. 9

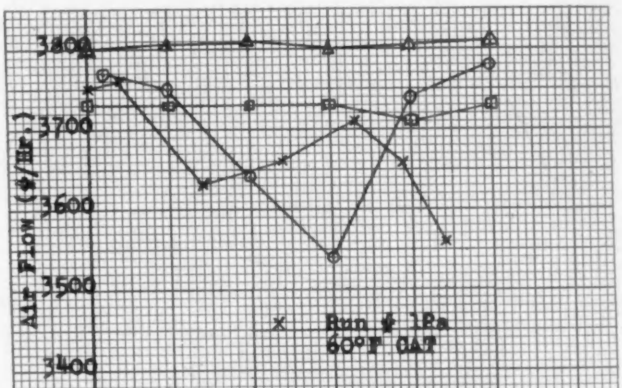
Table I – Summary of Engine and Carburetor Icing Tests

Run No.	Length of Run, min.	C. A. T., °F	Water Conc. (1)	Droplet Size 1-2-3 (2)	Carb. Air Flow lb./hr. Set.	Carb. Air Flow lb./hr. Min.	Throttle Angle	F/A Ratio Setting	F/A Ratio Max.	F/A Ratio Min.	Fuel Temp., °F	Initial Throttle Pres. Drop, in. Hg.	Time at near Constant Air Flow min.	Time during Decrease of Air Flow min.	Blower Case Metal Temp. °F Max.	Blower Case Metal Temp. °F Min.	Effect on Engine Operation (3) Air Flow	Effect on Engine Operation (3) F/A	Type Ice Formed (4) Impact Ice	Type Ice Formed (4) Refrigeration	
7	2.5	31.5	10 g/M^3	2	4000	1900	21.5	.070			68	7.75	0	2.5	32	19	2	1	2	-	Throttle frozen
8	13.5	35	"	2	4000	2700	21.5	"	.092	.069	69	7.78	11	2.5	36	24	2	1	-	2	F/a became richer
9	13.0	41	"	2	4000	2800	21.5	"	.072	.049	69	7.7	9.5	3.5	34	30	2	2	-	2	F/a became leaner
10	19.0	45	"	2	"	2660	"	"	.094	.072	69	7.6	12.5	6.5	34	30	2	1	-	2	F/a became richer
12a	26.0	59	"	2	"	3500	"	"	.072	.062	71	7.8	0	26.0	41	32	2	1	-	2	Ice reduced air flow immediately
12b	27.0	63	"	2	"	3550	"	"	.078	.062	73	7.73	0	27.0	42	33	2	1	-	2	" " " " "
12c	25.0	67	"	2	"	3800	"	"	.073	.072	74	7.91	25.0	0.0	42	33	1	0	-	1	Ice formation stabilized
12d	25.0	74	"	2	"	3720	"	"	.078	.075	76	7.8	25.0	0.0	47	38	0	0	-	1	" " "
13	30.0	40	75%	2	"	3930	"	"	.073	.072	50	7.85	30.0	0.0	43	36	0	0	-	1	" " "
14	30.0	39	100%	2	"	3880	"	"	.077	.074	71	7.85	30.0	0.0	24	14	0	0	-	1	" " "
15	55.0	39	5	2	"	3050	"	"	.076	.064	72	8.0	26.0	29.0	34	28	1	1	-	1	" " "
16	17.5	39	10	2	"	3100	"	"	.081	.063	72	8.5	2.0	15.5	42	35	2	1	-	2	
17	5.5	39	20	2	"	-	"	"	.073	.066	71	8.1	0.0	5.5	43	40	2	1	-	2	Ice formed too fast to obtain complete data
18	3.5	40	25	2	"	-	"	"	.065	.062	71	8.1	0.0	3.5	40	38	2	1	-	2	" " " " " " " "
20	32.5	34	100%	2	"	3810	"	"	.077	.075	71	7.9	32.5	0.0	34	24	0	0	-	1	
21	20.5	34	5	2	"	3060	"	"	.097	.077	70	7.9	2.5	18.0	35	20	1	1	-	1	Mixture became richer
22	7.5	35	10	2	"	3270	"	"	.086	.076	72	7.8	2.0	5.5	35	20	2	1	-	2	
23	2.0	33	20	2	"	2900	"	"	.097	.075	72	7.8	0.0	2.0	33	19	2	1	-	2	Mixture became richer
31	30.0	40	5	2	6000	5440	21.5	.070	.074	.053	73	12.05	0.0	30.0	35	22	1	2	-	1	
32	60	40	5	2	4000	2810	21.5	.070	.081	.072	74	8.0	18.0	42.0	42	26	2	0	-	2	
33	35	38	5	2	3000	2970	21.5	.070	.069	.070	75	3.0	35.0	0.0	38	23	1	0	-	1	
35	32	40	5	2	3000	2910	15.0	.070	.072	.070	63	10.9	32.0	0.0	41	28	0	0	-	0	
36	25	40	5	2	2000	1950	13.0	.070	.077	.062	66	12.75	5.0	20.0	40	27	0	0	-	0	

(1) Water concentration in % Relative Humidity or in grams of water per cubic meter of dry air in excess of 100% Humidity.
(2) Droplet size, 1, small, 2 medium and 3, large size droplets.
(3) Effect on engine operation, 0, none; 1, slight; 2 definite.
(4) Ice formation, 0, none; 1, slight stabalized formation; 2, unstable formation tending to build up.



■ Fig. 11 – Carburetor air flow versus time with constant moisture content of 10 g/ cu m – F/A = 0.070



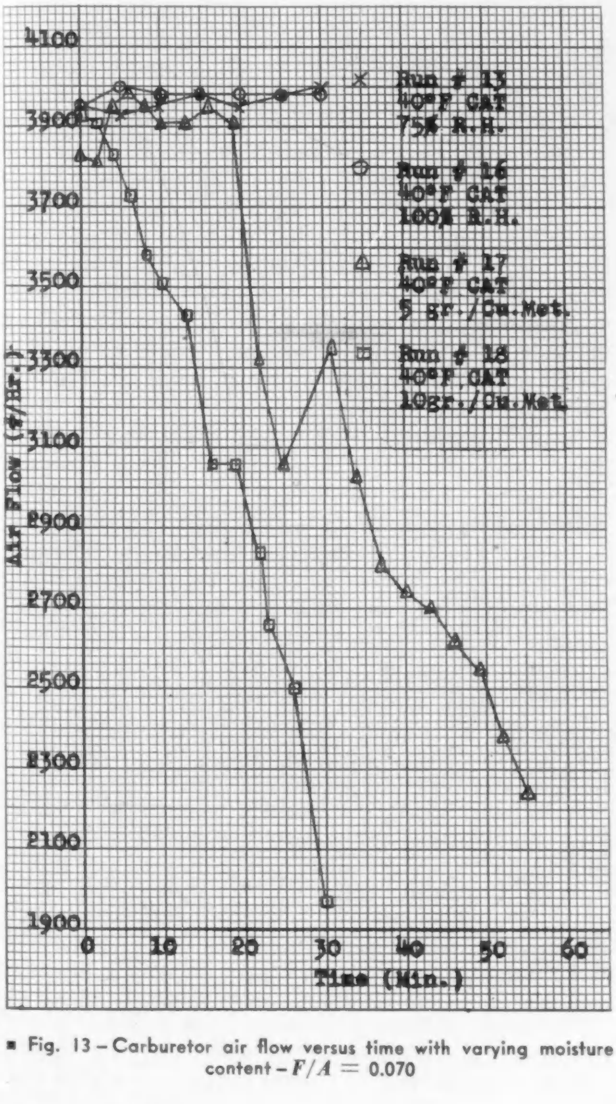
■ Fig. 12 – Carburetor air flow versus time with constant moisture content of 10 g/ cu m – F/A = 0.070

Preliminary Results

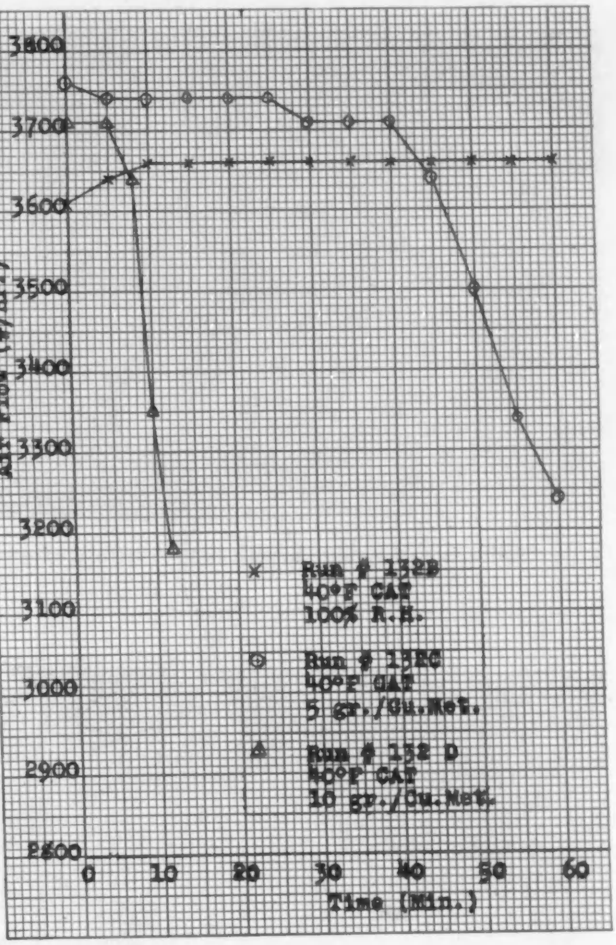
When the de-icing investigation was started, little was known about the actual mechanism of the building up of the ice formation. A study was made of this mechanism, first through windows cut in the sides of the adapter and later by taking the carburetor off at certain time intervals and examining the progress of the formation. A progressive series is shown in Figs. 2 to 7, taken under normal icing at cruising conditions, that is, 4000 lb of air per hr, 10 g per cu m rain density, 0.070 fuel-air ratio, and 20 deg throttle opening.

The series of pictures was taken during a total of 10 min of operation. Figs. 2 and 3 were taken at 1-min intervals and the others at 2-min intervals. There was no appreciable loss in air flow until the last 2 minutes of operation. This loss amounted to about 1200 lb per hr at the termination of the run. Fig. 8 shows the results of a run made under similar conditions where the carburetor was not removed until the ice was formed completely.

Fig. 9 shows a graphical record of the air flow taken during two runs at 5 and 10 g per cu m rain density. Fig. 10 shows the corresponding gasoline flow chart for the same runs. These charts serve to illustrate the character of the changes which take place and the relatively short times during which they occur. The time between the heavy radial lines is 30 min and that between the lighter lines is 7½ min.



■ Fig. 13 – Carburetor air flow versus time with varying moisture content – $F/A = 0.070$



■ Fig. 14 – Carburetor air flow versus time with varying moisture content – $F/A = 0.085$

Table 1 gives a summary of 23 runs made at the Bureau on the G-200 Wright Cyclone engine using the 1375 F Holley carburetor. These runs are divided into 5 groups.

The first group consisting of runs 7 to 10 has been plotted in Fig. 11 and shows the effect of a change in temperature on the icing time. Fig. 12 shows a group of runs made at higher temperatures with the same rain density as the first group. Fig. 13 shows some runs of the third group where the moisture content of the incoming air is varied. For comparison another group of curves is shown in Fig. 14 taken under similar conditions but with a richer mixture.

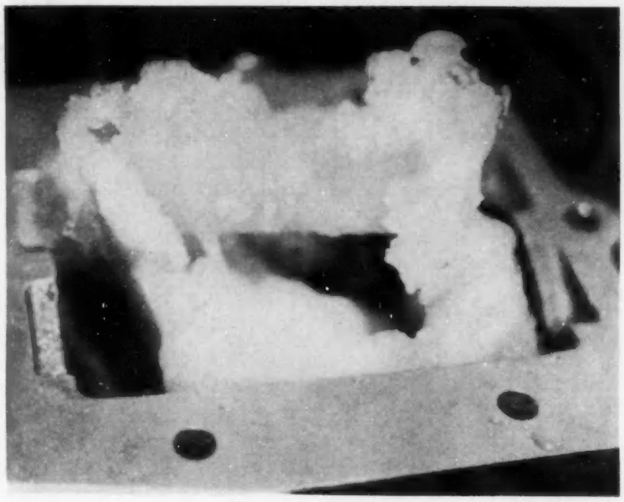
The conclusions formed at this point in the research program are as follows:

1. The rate of ice accretion is roughly proportional to the rate of water ingestion.

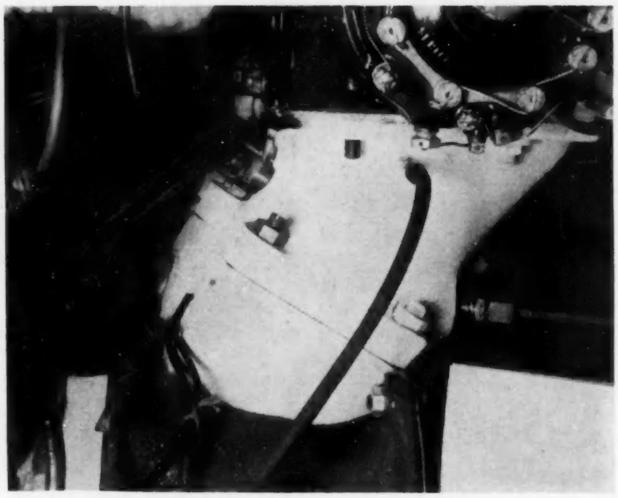
2. The rate of ice accretion increases with an increase in the fuel-air ratio.

3. The rate of ice accretion increases with a decrease in droplet size particularly below freezing temperatures.

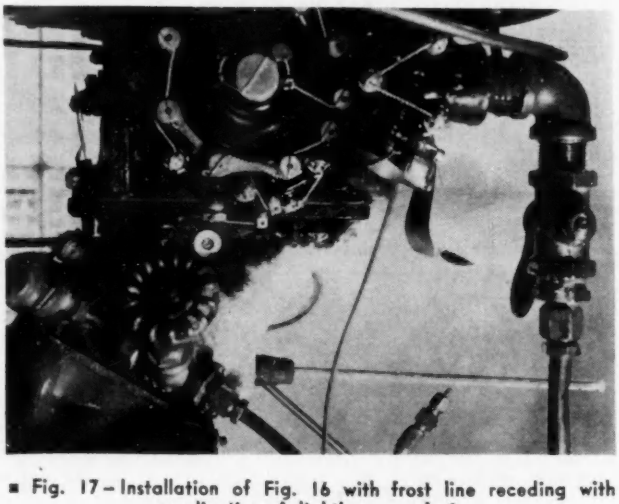
4. The most dangerous condition was found to be



■ Fig. 15 – Ice formation showing space between ice and adapter



■ Fig. 16 – Carburetor installation with outside of adapter completely frosted over



■ Fig. 17 – Installation of Fig. 16 with frost line receding with application of slightly warmed air

around 30 F air temperature with a small droplet size. With these conditions the rate of icing was so fast that it was impossible to stabilize the gasoline flow before the air flow had dropped off.

After several of these runs had been made it was noticed that there was usually a space between the ice and the adapter, partially shown in Fig. 15, which indicated that the bond between the ice and the adapter had been broken. Further investigation showed that the ice was not supported by a bond but by the shape of the adapter itself and the protuberances therein.

By experimenting with the outside temperatures, it was found possible to control the bond between the ice and the metal and the following interesting results were obtained: Fig. 16 shows the outside of the adapter completely frosted over and Fig. 17 shows the frost line beginning to recede with the application of slightly warmed air.

From these results it was concluded that tests should be conducted with a warm air blast against the adapter having a temperature and velocity equal to that found behind the cylinder baffles of an aircraft under normal operating conditions. It was concluded that the induction system could be made substantially ice-free if certain rules were observed in designing the system, that is:

1. The induction system below the carburetor should be so designed that there will be no change in the aspect ratio of successive sections such as to prevent the escape of an ice formation which has been loosened from the surface of the metal.

2. There should be a slight draught in the down-stream direction of the induction passage so that the ice may fall off readily from the metal walls.

3. There should be no nuts, bolts, studs, thermometer bulbs, thermocouples or other protuberances extending beyond the wall of the intake passage into the mixture stream which will hinder the removal of the ice formation.

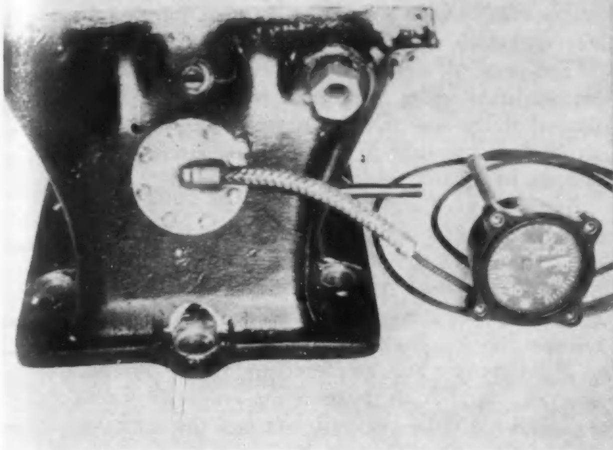
In studying the mechanism of the formation, it was concluded that either finning the adapter or even oil jacketing would reduce the thickness of the ice which could form if the design rules were followed.

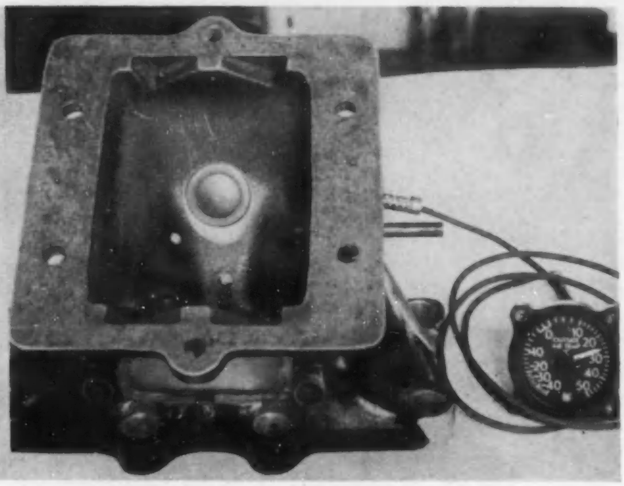
It has been suggested that, with the great amount of refrigeration available in the adapter, it should be possible to do away with the conventional type of oil cooler by diverting the oil return line through a jacketed adapter.

The recommendation regarding the elimination of thermometer bulbs which protrude into the induction system and pick up ice, brought up another problem. Many operators depend on adapter thermometers for mixture temperature indication. Furthermore, mixture thermometers seemed desirable for light plane operation during the winter months since they indicated the effectiveness of the air heating system. In order that distance-type thermometers might be used without introducing an icing condition into the intake passages, a flush-type thermometer bulb has been developed and the experimental model is shown in Figs. 18 and 19. The electric type of thermometer should be available for test shortly.

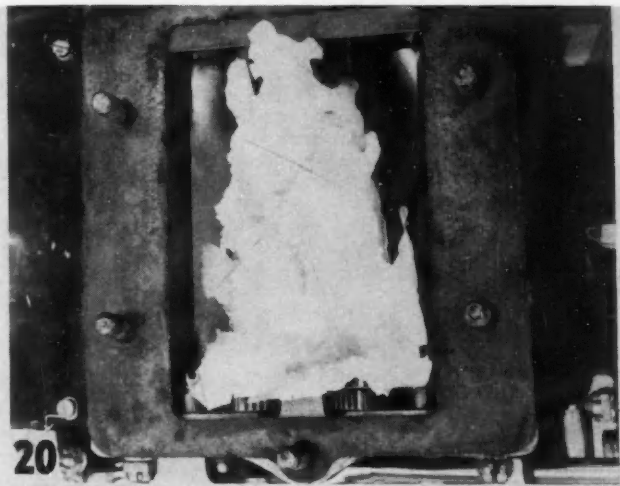
■ Other Types of Ice

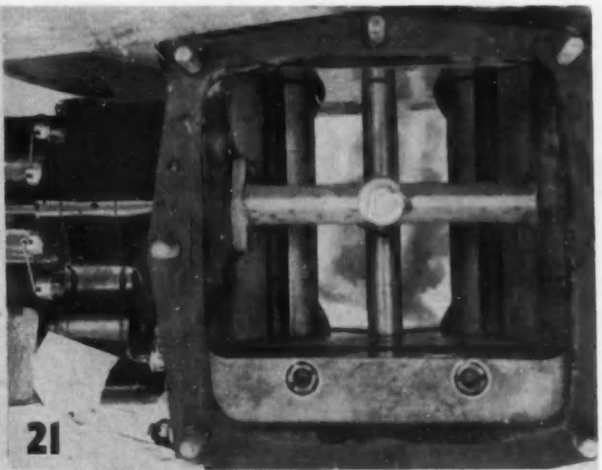
The transition from refrigeration ice to impact ice as the temperature is reduced is shown in the following figures: One formation taken at a lower temperature is shown in

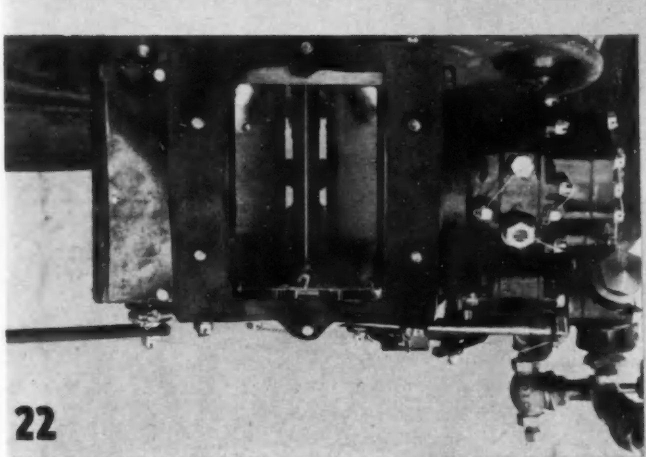


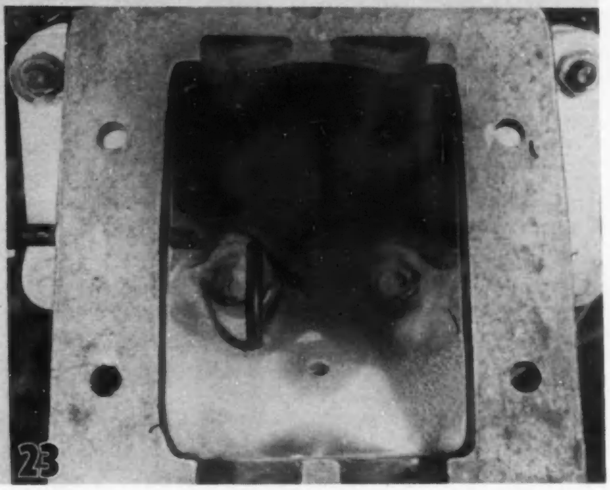


■ Figs. 18 and 19 – Two views of experimental flush-type thermometer bulb installation







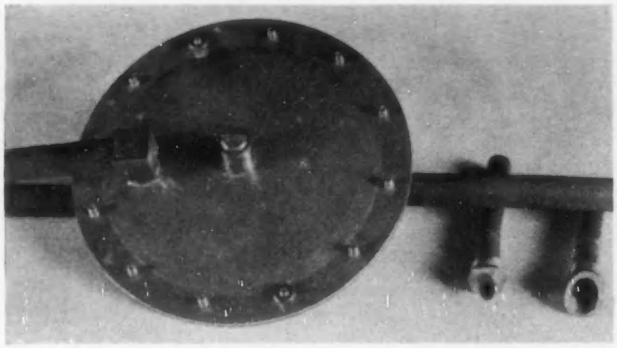


■ Fig. 20 – Formation of ice clinging to carburetor nozzle bar

■ Fig. 21 – View of same formation shown in Fig. 20, but looking down from the top of the carburetor

■ Fig. 22 – Carburetor ice formation at 30 F air temperature

■ Fig. 23 – Slight formation of ice in adapter under same conditions as carburetor formation of Fig. 22



■ Fig. 24 – Tube type of icing indicator

Fig. 20. This ice actually clung to the nozzle bar and did not form in the adapter at all. A view of this same formation looking down on top of the carburetor is shown in Fig. 21.

At 30 F air temperature and with the same conditions as before, icing was produced as shown in Fig. 22. Under these conditions it was impossible to move the throttle at all and the air flow was reduced from 4000 lb to 2800 lb per hr in about 3 min. Fig. 23 shows the small amount of ice formed in the adapter under these conditions.

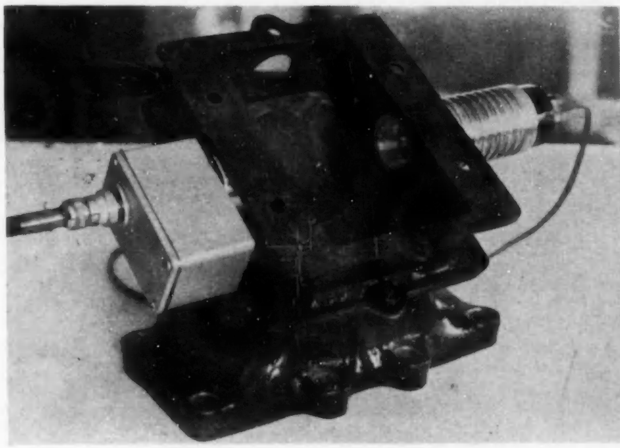
the aircraft. With the tube type of indicator, this delay was accomplished by finding a location for the tubes in the adapter which would give this effect.

The principal elements of the tube-type indicator are two impact tubes and a diaphragm valve. The smaller tube is located at the top of the adapter to register impact icing while the large tube is placed in the lower portion of the adapter to register refrigeration ice.

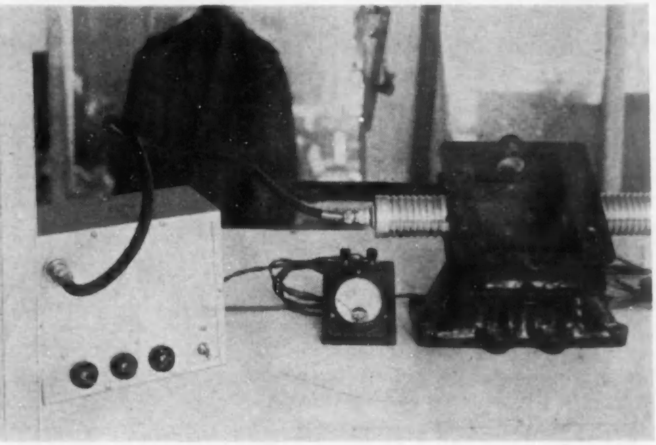
The photo-cell indicator is shown in Fig. 25. This instrument consists of a light source installed on one side of the adapter and the photo cell and one stage of amplification on the other. As ice forms in the adapter the light intensity at the photo cell is reduced. The delay can be regulated by moving the lenses back from the adapter wall.

The indicator and amplifier are shown in Fig. 26. In an actual aircraft installation the amplifier would be installed in the radio compartment and the indicator on the instrument panel. With this type of instrument, "wolf ice" is indicated by a fluctuation of the needle. A full-scale deflection of the needle indicates that, if the pilot should open his throttle wide, he probably would not obtain full power from his motors.

The problems of induction-system icing have been most interesting from a laboratory standpoint. Difficulty in duplicating results has shown us that many factors are contributing to the phenomenon which have not yet been taken into account.



■ Fig. 25 – Photo-cell indicator



■ Fig. 26 – Photo-cell indicator and amplifier installation

■ Icing Indicators

Since most of the corrective measures suggested involve design changes which cannot be embodied readily into present equipment, considerable interest has developed in the possibility of icing indicators.

Two indicators have been tested at the Bureau, one of the tube type using pressure differences for indication shown in Fig. 24 and the other a photo-cell type of indicator.

The main problem in ice indication is to delay the action of the indicator so that the normal coating of ice in the adapter will give no indication. This coating of ice, usually referred to as "wolf ice," does not affect the operation of

The apparatus used has been found to be very dependable and, as the investigation progresses, we expect to be able to control the variables with greater ease.

I want to take this opportunity to thank the industry for its splendid cooperation in supplying the Bureau with equipment and data necessary to carry on this investigation. Our requests have always been answered promptly and generously.

Our hope is to furnish information which will enable operators to reduce substantially the number of accidents and interruptions during the coming winter and, during the winter of 1942-1943, to eliminate all accidents from this cause.

DISCUSSION

Compares Results with Those of P&W Studies

—Victor J. Skoglund

Pratt & Whitney Aircraft, Division of United Aircraft Corp.

As a result of a request of the airlines at the Engineering Maintenance Conference of the Air Transport Association in July, 1940, Pratt & Whitney Aircraft also undertook an induction-system icing study in conjunction with the development of a new power-plant installation. That study was started at about the same time as the one reported by Mr. Kimball, and is described in the October, 1941, issue of the *Journal of Aeronautical Sciences* in a paper entitled: "Icing of Carburetor Air Induction Systems of Airplanes and Engines." The two studies have been coordinated through committee meetings and personal visits to the mutual benefit of both of the studies.

Mr. Kimball mentioned that heat is commonly used to prevent ice, but that it involves a loss of power. The relative importance of different factors involved in the use of preheat for de-icing is not generally understood. The following should be recognized:

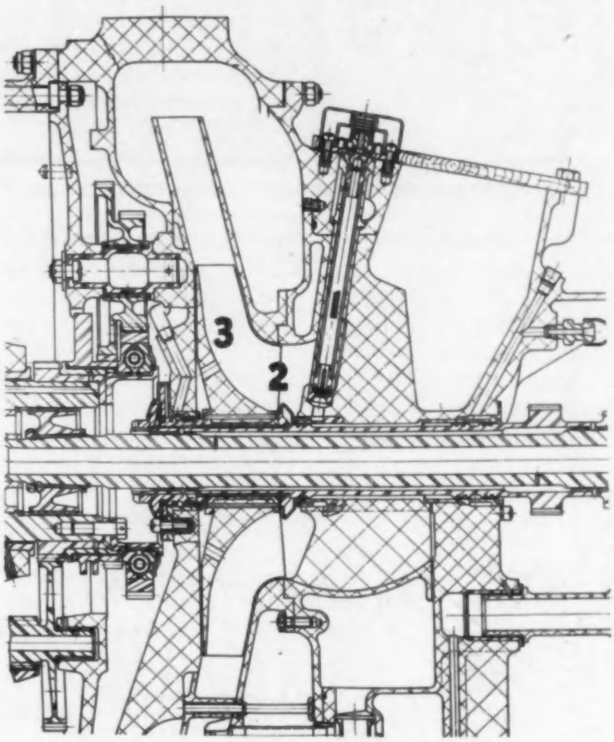
1. Carburetor air temperatures required for preventing or removing ice are not greater than temperatures encountered in the atmosphere of some localities.

2. There is no loss in available power until the critical altitude is approached since the throttle may be opened to compensate for any decrease in intake port density. Above critical altitude, the loss due to the decrease in density is so small that it is unimportant in airline operation except in emergencies such as single-engine climb. Under these conditions airplane speed is low, so that the effect of the loss in ram of the hot air is almost negligible.*

3. Some type of protected air inlet is necessary to eliminate impact icing hazards, so that a preheat system in conjunction with it involves little additional mechanical complication or weight.

In comparing the results of the subject and Pratt & Whitney studies, the following important features of our equipment should be considered:

1. A Bendix PD-12F3 pressure-type carburetor with two radically different fuel discharge systems was tested. The first type was the conventional Pratt & Whitney X-bar, illustrated in Fig. A. The second was the recently developed Spinner, illustrated in Fig. B. In this system the fuel discharge nozzle rotates with the supercharger impeller, the fuel being sprayed radially into the airstream at the impeller entrance. The Spinner installation is less susceptible to fuel evaporation ice than any other current system. This is the first public description of the Spinner because it has just been removed from the secret patent list.



■ Fig. B (Skoglund discussion) – Spinner fuel distributor (1) fuel pipe; (2) fuel slinger; (3) impeller

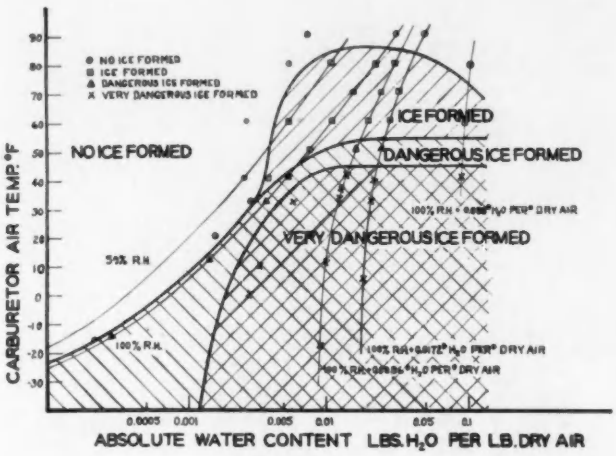
■ Fig. A (Skoglund discussion) – Pratt & Whitney X-bar fuel distributor

2. A complete running engine rear section driven by an electric motor was utilized.

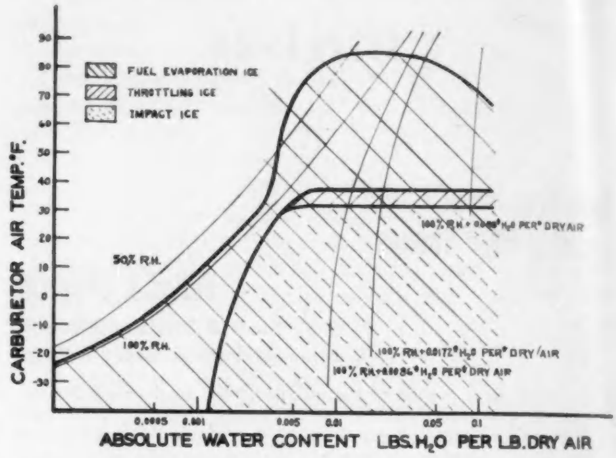
3. The carburetor was bolted directly to the intermediate case of the supercharger.

Before the present two laboratory studies, few people recognized the different ways in which ice may form in an induction system, the methods of preventing its formation, or the hazards of different kinds of ice. Three distinct kinds of ice, rather than the two mentioned by the author, are important. We defined impact ice as that which is formed with air temperatures below 32F from particles which originally existed in the free air in a liquid or solid state. Throttle ice, not mentioned by the author, was defined as that which is formed at or near the throttle due to the temperature drop which is produced by the increase in kinetic energy of the air at the restricted flow area of a partially closed throttle. This type of ice may form entirely independent of impact ice without fuel flow at carburetor air temperatures above 32F. Fuel evaporation ice was defined as that which is formed due to the cooling effect of the fuel evaporating into the carburetor air stream. The term "fuel evaporation ice" was used rather than "refrigeration ice" because it best describes the actual process.

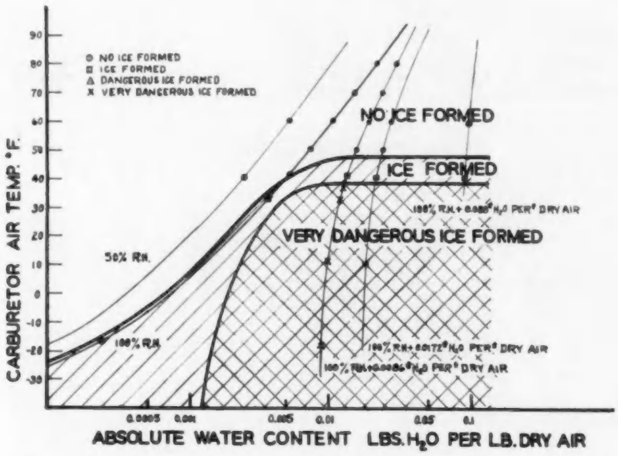
In our tests, the formation of ice was mainly controlled by temperature and water content of the carburetor air, the effect of all other variables being slight. Figs. C, D, E, and F indicate the effect of these two variables. Fig. C indicates limiting icing conditions with the X-bar. Fig. D indicates limiting icing conditions with the Spinner. Curves of Fig. E indicate a comparison between conditions under which no ice formed with the X-bar and Spinner. Fig.



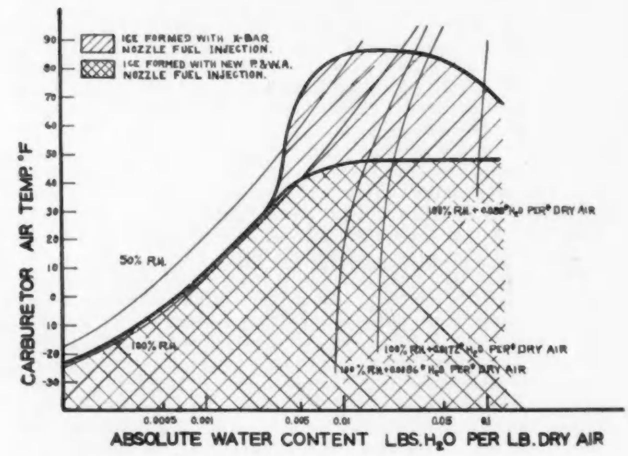
■ Fig. C (Skoglund discussion) – Limiting icing conditions, X-bar fuel distributor



■ Fig. F (Skoglund discussion) – Icing conditions for different types of ice



■ Fig. D (Skoglund discussion) – Limiting icing conditions, Spinner fuel distributor



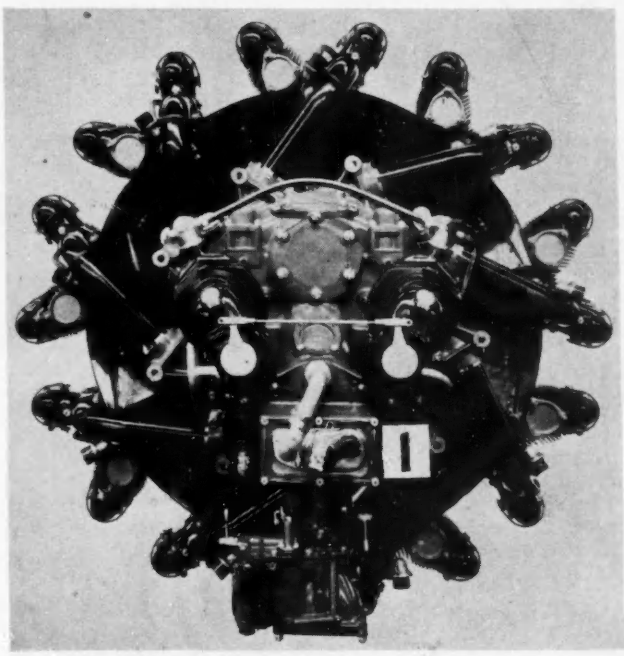
■ Fig. E (Skoglund discussion) – Comparison of icing conditions with X-bar and Spinner fuel distributors

F indicates the conditions under which the three different types of ice were formed with the X-bar.

Contrary to Mr. Kimball's results, drop size had only a slight effect on the rate of ice formation in our tests. Icing characteristics were almost the same where excess water over 100% relative humidity was injected with steam nozzles in one case and water spray nozzles in the second case for the same final carburetor air temperature and other conditions. A difference in drop size produced by the two methods was observed through windows in the air duct. The fact that heat-transfer conditions were not the same just below the carburetor might explain this discrepancy in the results of the two studies.

Ambient air temperature had little effect in our tests because carburetor metal temperatures were mainly controlled by conditions within the engine rear section.

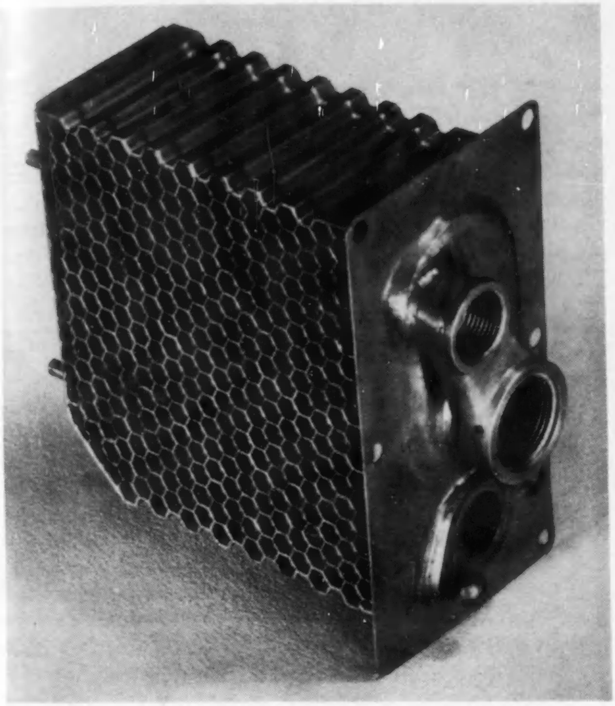
Mr. Kimball recommends that the aspect ratio of all sections below the carburetor be the same. This is practically impossible with current carburetors, and does not seem necessary since ice may be pre-



■ Fig. G (Skoglund discussion) – Mixture-cooled oil cooler on Wasp Junior engine. Cooler itself is shown at (1)

vented from jamming against the walls even though the aspect ratio is changing.

With regard to utilizing the cooling effect of evaporating fuel for cooling lubricating oil, Fig. G illustrates such a cooler installation used on a 1933 model of a Wasp Junior engine. Fig. H is a photograph of the cooler itself. Experience with these installations indicates that there is not sufficient cooling capacity for engines with rated powers greater than 500 hp. In any case, the space required



▪ Fig. H (Skoglund discussion) – Mixture-cooled oil cooler

for the heat-transfer surface makes it impractical to locate it below the carburetor on present high powered engines.

An ice warning device or an icing indicator is desirable in conjunction with adequate icing protection to eliminate all induction system icing hazards. Ice warning devices refer to instruments which indicate icing conditions before ice is formed, and icing indicators refer to instruments which indicate that ice has formed. Our tests indicated that ice which is dangerous varies in quantity and location. For that reason, an ice warning device is considered preferable to an icing indicator especially if the indicator operates on a localized effect, and since it is desirable to operate without any ice in the induction system.

Although considerable basic data have been obtained recently on induction-system icing, there are a large number of unsolved problems. It is hoped that the Special Subcommittee on Induction System De-Icing of the Power Plant section of the NACA with its excellent facilities at the National Bureau of Standards under the supervision of Mr. Kimball will continue its excellent work and that it will become the clearing house of all of those problems.

Diesel Engines for Trucks–Why and When

AS long as the diesel engine was and is used as a slow-speed heavy-duty engine in marine or industrial service, it operates on a reasonably close approximation to the ideal constant-pressure cycle. However, when the size was reduced and speeds were increased and, particularly, when the engine had required of it operation under widely varying load and speed conditions, a marked departure from this cycle occurred. This, of course, means that not only the compression but the maximum pressure are measurably greater than in the constant-volume engine. Fortunately, the variations in maximum pressures between full load and part load are not so great, but the necessary provision of bearings, rods and pistons to cope with these greater pressures develops a reciprocating weight which quite definitely limits the top speeds at which this engine can be operated.

When one asks himself why has the diesel engine excited so much interest the answers seem to fall into a pattern somewhat as follows:

1. The efficiency of the cycle is better, particularly at part-load, which results in lower total and specific fuel consumption.

2. Because fuel injection can be applied more easily, a fuel can be used which presently costs less.

3. The use of fuel injection makes possible an easier solution of the two-cycle principle and the use of a supercharger, still further accentuating (1) and (2).

The facts at the present time justify the opinion that diesel engines as currently developed are satisfactory for trucks. The time has not yet arrived when the advantages of the diesel engine are so outstanding that a complete replacement of the carburetor engine is imminent. The service in which the diesel engine can develop its advantages most effectively is in heavy-duty, long-distance hauling where an engine of large displacement can be used to advantage. Also successful installations have been made where loads were heavy and a big engine was desirable and, while total mileage was not large, total fuel consumption was large. What this means is that the economies in the diesel cycle develop into profits in this type of service.

We may not be justified in the opinion that the smaller sizes of engine, suitable for the large volume of transport, have not yet reached as satisfactory a stage of development as the larger engine. However, experience indicates that further evidence must be obtained before we can be sure that this is not so. It should be understood that what is referred to does not refer to design or construction alone, or even primarily, but to all of the factors which are inherent to the problem.

In opening this article it was stated that the comparison of the relative merits of the carburetor and self-ignition engine is one of those subjects that is the source of perennial debate. We are not so optimistic as to feel that this discussion contributes in any important degree to forming a conclusion to the debate on this subject. With some people, the conclusion has already been reached. Unfortunately, the conclusions are not the same. One group feels that diesel engines are the only kind which have any value, and the other, having made an inadequate and unsatisfactory trial, thinks they are all bad. Obviously, neither of these views is correct.

The attempt has been made to point to the field where successful operation has been obtained and to some difficulties encountered in extending this field. One thing seems more certain today and that is that reliability essential to success is now available in several engines on the market. When this point is reached, that which remains is the certain establishment of profit. As it becomes evident that this can be obtained, the diesel engine will progress through the various branches of truck transport. As this process can be viewed at present, it will take quite a while. As a matter of fact, it would, in our opinion, be a bold prophet who would attempt to assign a more definite term.

Excerpts from the paper of the same title by B. B. Bachman, vice president of engineering, The Autocar Co., presented at the Metropolitan Section of the Society, New York, N. Y., Nov. 13, 1941.

Facts and Fallacies of

EVERY engineer is taught how to solve for stress in loaded members and, after becoming reasonably proficient at stress problems, he is advised that it is a pretty good idea to assume that his answers are from 50 to 90% wrong. Of course, he is not advised in these words or with this emphasis. More elegantly, a 50% probable error becomes a factor of safety of 2, a 90% probable error becomes a factor of safety of 10; and thus, the greater the doubt, the safer we feel. The fact is that no means are yet available for reliable determination of stresses in dynamically loaded, highly stressed machine parts. Stress calculations are entirely inadequate except for a few special cases for which empirical formulas have been constructed from extensive service and experimental data. Photo-elastic analyses can be made to reveal major stress patterns but do not provide useful quantitative values of stress. Extensometer readings, brittle lacquers, and so on, likewise can show major stress patterns but fail to provide reliable quantitative stress values. Even if true stress could be found, the permissible working stress cannot be judged from fatigue tests on laboratory specimens.

■ Heavy Machines Overdesigned

In the design of heavy machines and static structures, the shortcomings of our means for determining stress are overcome by overdesign, commonly considered as factors of safety. For low-volume production machines in which weight is relatively unimportant, this practice is well justified because the cost of designing to low stress tolerance would far exceed the gain from savings in material and weight. The case is very different when we consider machines in which weight is all-important, as in airplane engines or where weight and cost must both be considered as in automobiles and other mass-production items. In such machines it is necessary that each unit of material shall do the maximum of work consistent with reasonable cost, and it is in the design of such machines that the inadequacy of our means of stress determination and our ignorance of fatigue strength of materials are most keenly felt.

In the design of light machines where we cannot seek safety in oversize parts but must design as exactly as possible to the required strength, we frequently find that an experienced designer can estimate the required dimensions of a part more accurately than they can be calculated by textbook methods. Nevertheless, stress calculations must be made as first approximations in the design of parts for a new machine. Such calculations usually follow conventional methods but, instead of applying arbitrary factors of safety, we are guided by accumulated experience in the form of limiting stresses which vary with the kind of machine part that is being considered. For example, as rough measures, automobile crankshafts may be stressed, by calculation, to 20,000 psi; connecting rods will tolerate calculated stresses of 40,000 psi; valve springs do somewhat better, approximately 90,000 psi; disc clutch springs are good for 180,000 psi; while other forms of clutch springs may safely carry 600,000 psi calculated stress. These few examples are given to illustrate that calculated stresses are, in themselves, meaningless and are of value only when they are interpreted in terms of experience. At best, such calculations are little better than guesses and serve only as starting points. Each part must be tested thoroughly by all available means and modified as needed before it can be released for production which means only that our experience data are not, in most cases, sufficiently organized to permit the construction of reliable empirical formulas.

IT is in machines where weight is all-important, such as in airplane engines, or where weight and cost must both be considered, as in automobiles, that the inadequacy of our means of stress determination and our ignorance of fatigue strength of materials are most keenly felt, the author of this paper believes. Calculated stresses, he contends, are in themselves meaningless and are of value only when they are interpreted in terms of experience. Although conceding that the accuracy of stress data from photo-elastic and extensometer readings is usually greater than that obtained from the most involved mathematical analysis, he shows by fatigue tests that these methods are far from reliable.

In the major part of this paper, S-N diagrams

The accuracy of stress data from photo-elastic and extensometer readings is usually greater than can be obtained from the most involved mathematical analysis, but that they are far from reliable can be shown easily by fatigue tests. Two specimens of identical material, heat-treatment, and dimensions will show identical stress when measured by photo-elasticity or by extensometer; yet these specimens may vary widely in fatigue strength, depending upon minute differences in surface finish or internal stresses. Since internal stresses are often unavoidable due to processing operations, such as machining, heat-treating, straightening, or grinding, and since surface finishes vary all the way from rough forgings to lapped or honed surfaces, there is little reason to expect accuracy from extensometer readings and even less for photo-elastic tests since these specimens

[This paper was presented at a Detroit Section Meeting of the Society, Detroit, Mich., Oct. 13, 1941.]

[1] See *Metal Progress*, Vol. 32, February, 1941, pp. 202-206: "Improving Engine Axles and Piston Rods," by O. J. Horger and T. V. Buckwalter.

STRESS DETERMINATION

by J. O. ALMEN
Research Laboratories Division, General Motors Corp.

must be free from internal stresses and must be made of another material.

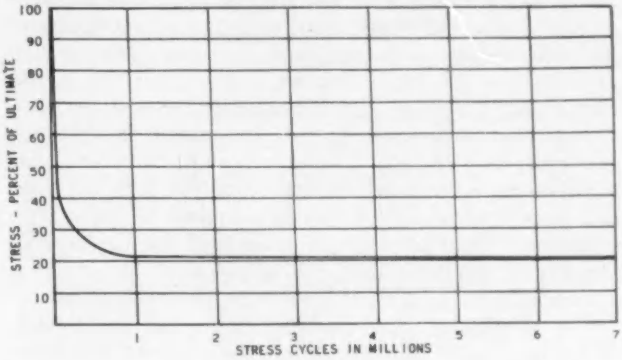
Photo-elastic and extensometer readings are measures of elasticity in which the changes in dimensions are the statistical average of all of the material involved in the measurement. Fatigue tests provide a strength measure of the weakest portion of the material involved, usually at the

with linear and logarithmic coordinates; combinations of these coordinates; and three-dimensional coordinates, are discussed. This discussion brings out the effect on fatigue strength of varying degrees of stress concentration; of surface treatment of the test specimens; of stress range; and presents considerable fatigue data on ball bearings. It is shown that, when fatigue-test results are run on a large number of commercially identical parts over a sufficiently large load range, the scatter of the test points, when plotted on logarithmic coordinates, falls within a well-defined pattern which tends to radiate from a point at high stress and low number of stress cycles and to diverge to a broad band at low stress and high number of stress cycles.

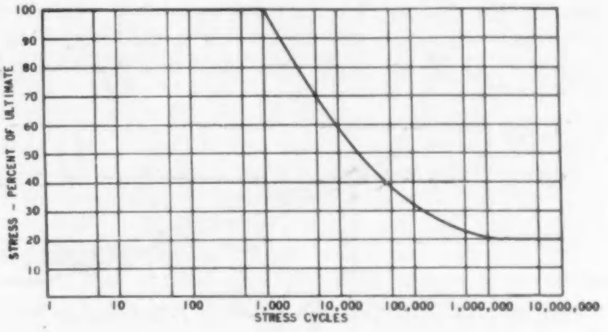
surface, even though it be sub-microscopic in size. Obviously, we cannot expect agreement between fatigue measures of stress and the stress readings obtained from elastic measurements.

This should not be construed to mean that stress-measuring devices should not be used. Photo-elastic tests are particularly useful in the classroom to demonstrate, qualitatively, the increase in stress at fillets, notches, holes and other section changes. Extensometer measurements have similar though somewhat broader utility since readings may be taken from actual machine parts to show the effect of section changes. These readings can sometimes point the way to alterations in design for more uniform stress distribution, providing that the manner of load application in service is known and not assumed, but they should be regarded as qualitative and not as quantitative measures of stress when applied to cyclically stressed parts.[1]

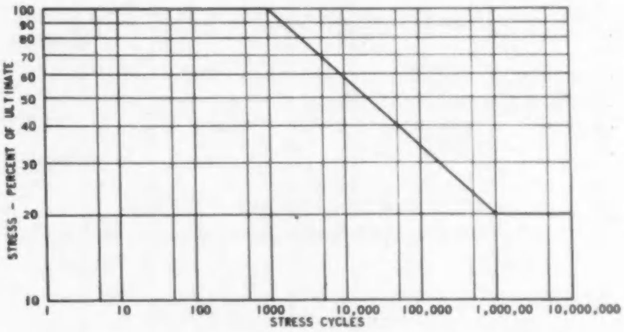
Fatigue test data as reported by many books and papers are not directly applicable to the design of light-weight, high-output machine parts because, with few exceptions, they assume: (1) that stress can be calculated; (2) that a machine part must be stressed below the fatigue endurance limit to be successful; (3) that laboratory test specimens are representative of a material when that material is used



■ Fig. 1 – Typical fatigue diagram plotted on linear coordinates



■ Fig. 2 – Typical fatigue diagram plotted on linear ordinates and logarithmic abscissa

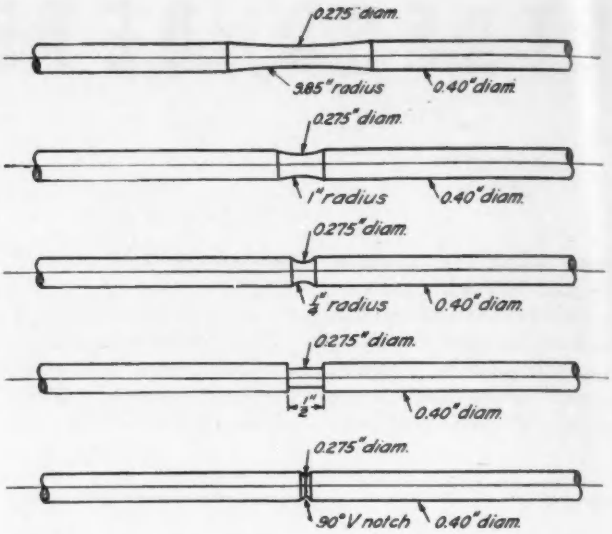


■ Fig. 3 – Typical fatigue diagram plotted on logarithmic coordinates

in a machine part; and (4) that a representative fatigue curve can be constructed from a dozen or less specimens.

Fatigue data are usually presented in graphs, called S-N diagrams, in which stress is plotted on the vertical axis and stress repetition or cycles are plotted on the horizontal axis. These graphs are variously constructed on linear coordinates, logarithmic coordinates or on linear ordinates and logarithmic abscissa. The linear coordinate plot shown in Fig. 1 is infrequently used in fatigue studies, notwithstanding the fact that it presents a clearer quantitative picture of the abrupt loss of strength of material when subjected to cyclic loads. As a working chart, however, this plot is inconvenient as will become apparent. The majority of fatigue workers prefer to use the linear ordinates and logarithmic abscissa scales since this arrangement contracts the N scale sufficiently to represent data over a large range of life. Fig. 2 shows the curve of Fig. 1 as it appears on such coordinates. The third form, less frequently used, is preferred for this paper because, as shown in Fig. 3, the fatigue curve of Figs. 1 and 2 appears substantially as a straight line when plotted on logarithmic coordinates. However, since there is no known way to determine true stress for machine parts, the stress scale in S-N diagrams can, at best, show only values proportional to stress. For specimens of varying shape, it is doubtful that the proportionality of the stress scale will be the same for any two specimen forms even though they are otherwise identical.

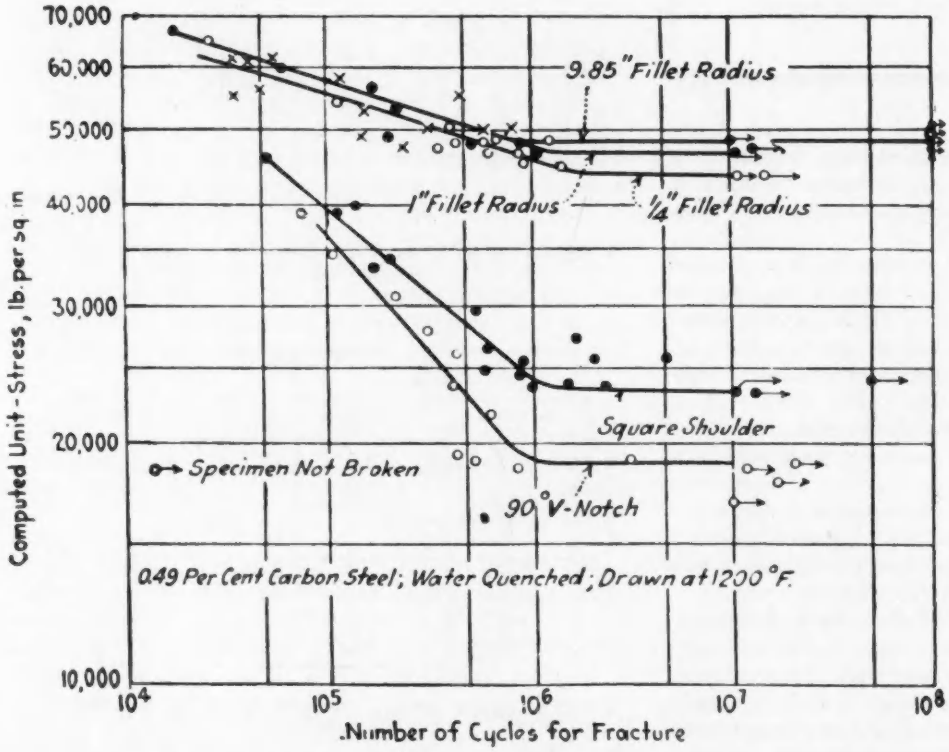
The assumption that machine parts must be stressed below the fatigue endurance limit has led most investigators in this field to test their specimens at stresses near this limit; that is, at stresses so low that the specimen will endure for an infinite number of stress repetitions without failure. There is, therefore, very little data from which to construct the characteristic curve of failure of specimens in the finite life region of fatigue – that is, that part of the diagram lying between one stress cycle and one million stress cycles. Many machine parts are operated at maximum stresses above the fatigue endurance limit and failure, when failure occurs in such parts, is due to a relatively small number of stress cycles at maximum stress. Table 1 shows the life requirements at maximum stress of a few typical machine elements as determined by service experience.



■ Fig. 4 – Specimens for study of effect of shape on endurance limit

These life requirements are fixed by the abusive type of user and are sufficiently high so that failure rarely occurs during the lifetime of the automobile except in the case of an accident. Obviously, to design such parts for infinite life at maximum stress would be economically wasteful.

The preferred laboratory fatigue test specimen is prepared very carefully to avoid all surface imperfections, abrupt section changes, internal stresses, and so on. This care is considered necessary because the investigator usually is interested in the properties of the material undergoing test and he naturally seeks to eliminate all factors that would tend to obscure these inherent properties. There can be no objection to this procedure as it refers to the test specimens, but the data thus obtained can have little bearing on the fatigue characteristics of machine parts made from the same material and given the same heat-treatment in which surface irregularities, abrupt changes in sec-



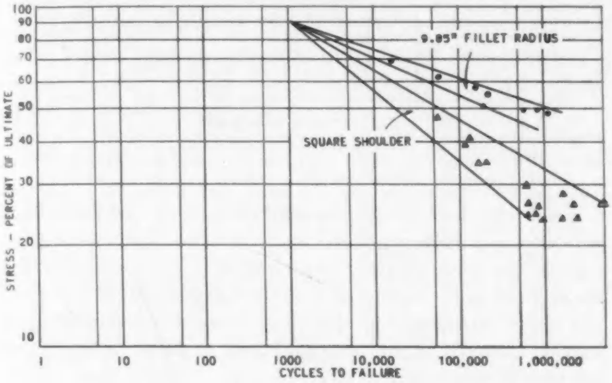
■ Fig. 5 – S-N diagrams for specimens of different shapes as shown in Fig. 4

Table 1 – Life Requirements at Maximum Stress in Cycles

Automobile rear-axle gears	100,000
Automobile transmission low gear	100,000
Auto chassis springs	100,000
Automobile transmission second gear	300,000
Truck rear axle	500,000
Bus rear axle	1,000,000

tion, and internal stresses are almost always present.

Many laboratory fatigue tests have been reported from specimens having varying degrees of surface roughness and various notch forms. These data are important in that they show very different characteristics from the smooth-surfaced uniform-section specimens. Fig. 4 shows a series of fatigue specimens used by Moore and Kommers[2] to determine the effect on fatigue of varying degrees of stress concentration. The resulting fatigue curves, plotted on logarithmic coordinates, are shown in Fig. 5. The authors



■ Fig. 6 – Probable form of scatter band of specimens 1 and 4, Fig. 4, if a large number of specimens had been tested

compare these specimens on the basis of calculated stress at the fatigue endurance limit; that is, the stress at the "knee" where the curve becomes horizontal. However, as stated before, our interest is in the finite life region of the diagram; that is, in the characteristic of the curve lying to the left of the "knee." Observe that, as the severity of the specimen section is increased the slope of the curve increases, and that the curves, if extended left-ward, tend to cross one another. Fatigue curves of machine parts, no matter how well finished or how carefully rejected for detectable flaws, almost invariably show steeper slopes than are shown by well-finished fatigue specimens and, therefore, presumably the fatigue strength of a material as determined by ideal test specimens is not obtainable when that material is formed into a machine part. Permissible stress at the fatigue limit of a machine part may be less than 10% of the ultimate strength of a material, whereas laboratory test specimens may indicate 50% or more as obtainable. The differences in slope of fatigue curves suggest that this characteristic promises a way whereby we may eventually greatly improve our accuracy in determining the strength of machine parts. This is now being done in rating the load capacity of ball bearings, roller bearings, and certain gears.

Published fatigue curves without exception are based on an insufficient number of test specimens. The lines plotted in Fig. 5 are intended to represent the averages for the specimens tested. Note the wide scatter of the test points and the increasing scatter of the points as the slope increases. Note also that, generally, the scatter decreases toward the left of the diagram. The significance of this scatter is not apparent in the diagram due to the limited number of test points, there being an average of only 12 failed tests for each type of specimen. Fatigue data should be regarded as mortality data, particularly in the finite life region of the diagram. We would have little confidence in life-insurance policies for which the premiums were based on mortality data from a dozen random individuals. Likewise, we cannot hope to establish the fatigue characteristics of specimens or of machine parts from a dozen odd tests.

The scatter of test points is due to unavoidable differences in test specimens no matter how carefully they are made and, since these differences constitute varying degrees of stress concentration, the fatigue line representing the poorest specimens should lie on a steeper slope than the fatigue line representing the best specimens. This is for the same reason that the average slopes of the specimens shown in Fig. 4 increase with the severity of the stress concentration as shown in Fig. 5. The test points for any group of specimens would, therefore, be expected to lie within a scatter band diverging from the region of high stress.

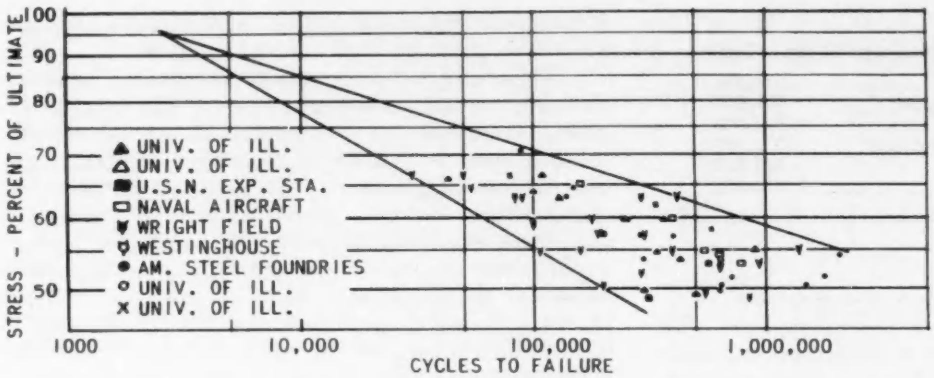
If a sufficient number of specimens had been tested, and if the stress scale proportionality were the same for all specimens, it is probable that the sloped lines in Fig. 5 would all tend to converge toward a point in the vicinity of 1000 cycles and 90,000 psi, somewhat as indicated in Fig. 6 in which the fatigue slopes of specimens 1 and 4 of Fig. 4 are shown as converging bands rather than lines. This region of intersection is suggested because the ultimate strength of the material tested by Moore and Kommers is approximately 95,000 psi and, obviously, if the stress scale is correct for each type of specimen, they will all have approximately the same strength at one stress cycle. The point of intersection would probably be at a considerable number of stress cycles because the ductility of the material permits adjustment of stress through yield, thus reducing the influence of local highly stressed points. For very brittle material, the intersection point of the fatigue curves for the type of specimens shown in Fig. 4 would probably be near the ultimate strength and nearer one cycle of stress.

There are not now available sufficient data on any specimens to complete a group of fatigue diagrams to the region of intersection. Knowledge of the characteristics of fatigue curves at high stress would be valuable in industry since it would facilitate greatly interpretation of fatigue tests on machine parts. Such tests could be evaluated in terms of the slope of the fatigue curve which would also give a clue to the actual stress, if desired, in the part being tested.

Fig. 7 shows an S-N diagram replotted from data contained in an ASTM Research Report[3]. By combining all the data for heat-treated SAE 4340 steel in a single S-N diagram, we have a total of 59 failed points which roughly outline a scatter band of the general form suggested in Fig. 6. This report also gives data on 54 failed points for

[2] See "Fatigue of Metals," by Moore and Kommers, McGraw-Hill Book Co.

[3] See Report of the Research Committee on the Fatigue of Metals, ASTM Meeting, June, 1941.



■ Fig. 7 – Re-plot of data reported by the ASTM Research Committee on Fatigue of Metals

■ Fig. 8 (below) – Fatigue properties of metals used in aircraft construction – 4134 steel

low-alloy steel (HT 50) which gives a less satisfactory scatter band since several points are reported as tested far above the ultimate strength of the material and the data are, therefore, doubtful.

Published data on fatigue of metals contain numerous tests showing the same general trend of increasing slope with increasing stress concentration, whether due to differences in specimen shape, specimen size, mechanical working of specimen surface, surface coatings, fillet radii, surface finish, or to variations between identical specimens. This tendency toward convergence is often not apparent in the published curves because the investigators have plotted their data on linear ordinates and logarithmic abscissa and always there are insufficient test points. The following diagrams copied from published papers have, when necessary, been replotted on logarithmic coordinates to the same scale as used by Moore and Kommers for the sake of uniformity, in which the stress scale is four times the scale of stress repetitions. The slopes of the curves are calculated as the measured horizontal distance multiplied by the scale ratio divided by the measured vertical distance. $\frac{(\text{Abscissa} \times \text{Scale Ratio})}{\text{Ordinate}}$ Strictly speaking, this value is the reciprocal of the slope, but it is used for convenience.

Oberg and Johnson[4] report a comparison between polished and notched specimens shown in Fig. 8 with results similar to the experiments by Moore and Kommers, Fig. 5.

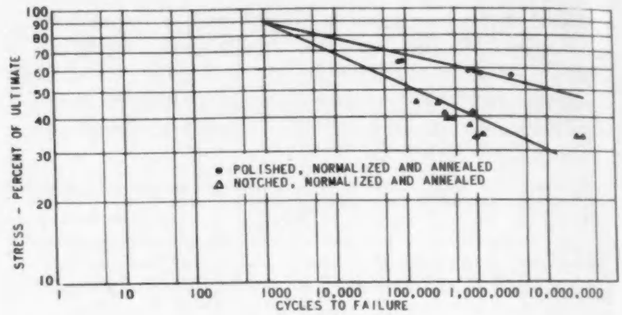
Surface treatment of the test specimens other than the degree of smoothness has a marked effect on fatigue strength. Horger and Maulbetsch[5] compared normal well-finished specimens with specimens that had been subjected to a rolling operation which introduced compressive stresses in the surface layer with the results shown in Fig. 9. Since the rolled specimens were pre-stressed in compression, the subsequent tension stresses during the test were reduced; hence the difference in the slope of the curves for the two types of specimens. Since this treatment should be ineffective in a tensile test, the lines should converge as shown.

Swanger and France[6] made tests on zinc-coated specimens by which they demonstrated loss in strength due to the zinc coat. Fig. 10 shows the increase in slope of the curve for the coated specimens and the convergence of the curves toward the left.

[4] See ASTM Proceedings, Vol. 37, Part II, 1937, p. 199: "Fatigue Properties of Metals Used in Aircraft Construction at 3450 and 10,600 Cycles," by T. T. Oberg and J. B. Johnson.

[5] See ASME Transactions, Vol. 58, 1936, pp. A-91-A-98: "Increasing the Fatigue Strength of Press-Fitted Axle Assemblies by Surface Rolling," by O. J. Horger and J. L. Maultbetsch.

[6] See ASTM Proceedings, Vol. 32, Part II, 1932, p. 439: "Effect of Zinc Coatings on the Endurance Properties of Steel," by W. H. Swanger and R. D. France.



From the foregoing, it seems reasonable, as a working hypothesis, to assume that, except possibly for very ductile metals, the slope of the fatigue curve may be considered a measure of true stress and that the fatigue curves for varying stress concentrations converge toward a point near the tensile strength of the material and at some considerable number of stress cycles.

The application of this hypothesis to the fatigue strength of machine parts leads to some important implications. A large variety of machine elements are being tested constantly for relative durability in the laboratories of industries engaged in the manufacture of light-weight, high-output machines. In most cases, these fatigue tests are intended to compare one design, material, or process with another design, material, or process. It is axiomatic that nothing can be learned in regard to limiting loads except through tests to destruction. Therefore, the fatigue tests for practically all parts are run to failure and the comparison is made on the number of stress cycles at constant load that each part will withstand. This procedure is followed regardless of whether, in practice, the part in question is stressed below the fatigue limit or whether it is a part requiring relatively short life at maximum stress.

This method of evaluating test results is subject to serious error for several reasons. If it is true that fatigue curves radiate from a point in the high stress region, it is obvious that comparisons of specimens cannot be made on a percentage basis only, since the percentage difference will vary all the way from zero to infinity depending upon the load that is applied during the test. Furthermore, since the scatter band for each test part should also radiate from the same point as in Fig. 6, the width of the band in terms of life may be several hundred percent and, unless a considerable number of tests are run for each part, there is no assurance that whatever life difference is found is real or just the chance location of those particular test points within the scatter band. It is easily possible that the better

design, material, or process will apparently rate lower than the poorer design, material, or process if an insufficient number of tests are made.

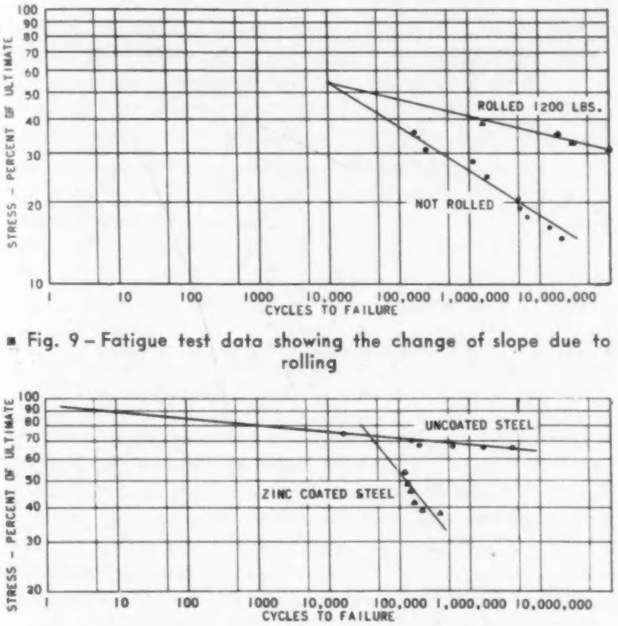
It is possible that the average fatigue curves for two materials having different tensile strengths and yield points will cross at some point in the finite life region due to differences in sensitivity to stress raisers. In such cases, life comparisons may be positive for one material at one test load and negative for the same material at another test load. It is evident, therefore, that true comparisons can be obtained only through fatigue tests on a sufficient number of parts at varying loads to outline the slopes of the scatter band limits. While this may appear to be an impractical requirement, it is not so difficult as it seems. It is only necessary that the results of the present routine tests be accumulated on a fatigue diagram and, in a relatively short time, fatigue curves and their scatter bands will be available for a large variety of machine parts.

Only occasionally are fatigue tests on machine parts run at various loads but, in the very few cases where data from a reasonable number of such tests run at sufficiently large load range are available on commercially identical parts, a reasonable number being one or two hundred, we find that the scatter of the test points when plotted on logarithmic coordinates falls within a well-defined pattern which tends to radiate from a point at high stress and a low number of stress cycles and to diverge to a broad band at low stress and high number of stress cycles just as was suggested in Fig. 6. This pattern is clearly shown in Fig. 11 which is a fatigue diagram of about 200 complete automobile and truck rear axles of various makes and sizes. The stress scale shown in this diagram is not true stress but is believed to be proportional to true stress. The axles were tested at loads to produce failure of one or more pinion teeth through the range of from 7000 cycles to 1,000,000 cycles. The scatter of the test points is due to variations in one or more of the many variables that are always present in commercially similar parts, such as internal stresses, fillet radii, cutter scratches, bearing, shaft and housing deflections, warpage in heat-treatment, and so on. The slope of the average durability line, calculated as the horizontal distance divided by the vertical distance measured on logarithmic coordinates is approximately 7, while the slopes of the upper and lower limits of the scatter band are, respectively, 9 and 5. The intersection point at the left of the diagram should lie near the ultimate strength of the material (approximately 300,000 psi) which, if proved, would supply us with a true measure of stress for the entire diagram. The diagram of Fig. 11 is not ideal as a proof of the scatter band or the intersection point, since it includes a variety of axles made from various alloy steels variously heat-treated for which the stresses were calculated by an empirical formula.

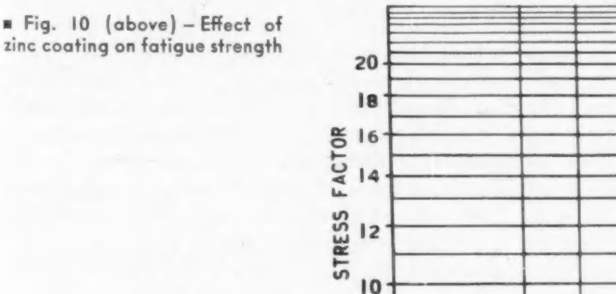
Satisfactory determination of the characteristics of the scatter band would require a large number of fatigue tests on one form and size of specimen, made of one type of material similarly heat-treated and tested to produce failure over a range of stress repetitions from as near a single cycle of stress as possible to the fatigue limit. Data approaching these requirements have been accumulated by the various ball and roller bearing manufacturers but very little has been published. However, fatigue data on ball and roller bearings need not in all particulars agree with fatigue data on other forms of machine parts since failure of roller bearings originates below the surface of the material. Surface influences, which play so important a part in fatigue of ordinary machine parts, are, therefore, absent in roller bearings which would be expected to influence the permissible stress and possibly the form of the scatter band. The scatter band as reported by Macauley[7] and by Styri[8]

[7] See *The Automobile Engineer,* Vol. 13, July, 1923, pp. 213-223: "The Endurance of Ball Bearings," by A. W. Macauley.

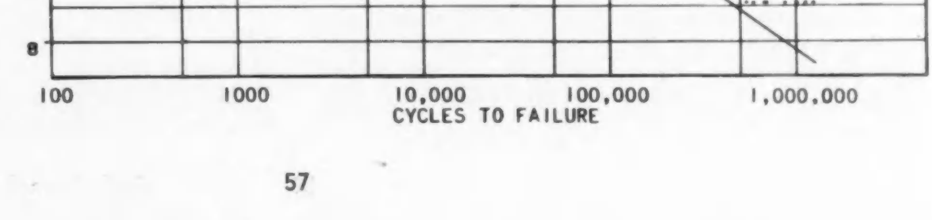
[8] See *Mechanical Engineering,* Vol. 47, June, 1925, pp. 490-492: "General Properties of Ball Bearings," by Haakon Styri.



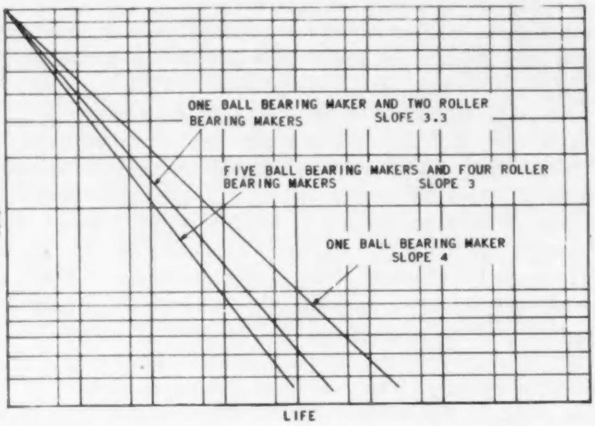
■ Fig. 9 – Fatigue test data showing the change of slope due to rolling



■ Fig. 10 (above) – Effect of zinc coating on fatigue strength



■ Fig. 11 – Scatter band for fatigue tests of 196 spiral-bevel rear-axle gears



■ Fig. 12 – Characteristic fatigue curve slopes, load versus life, as obtained from catalogs of ball and roller bearing manufacturers – In this diagram only the slopes are significant

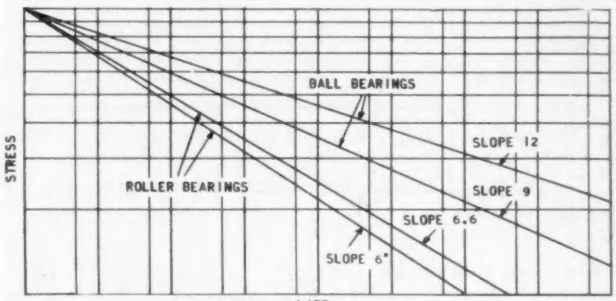
is parallel to the average life curve throughout the life range shown. The catalogs of ball and roller bearing manufacturers contain life and load data from which fatigue diagrams may be constructed but, although these diagrams reveal some very interesting facts, they do not supply us with scatter bands or show whether a logarithmic fatigue curve is, substantially, a straight line up to the ultimate strength of the material from which the bearings are made. Fig. 12 is a logarithmic diagram of load (not stress) versus life taken from the most recent catalogs of a number of leading ball and roller bearing manufacturers. All the lines, representing the various makes of bearings, are arbitrarily drawn from a point at the left of this diagram since we are now only interested in their slopes to show the similarity of the experience for all makes of bearings. The slopes for these lines are 3, 3.33, and 4.

It has been shown by Hertz and others that the stress in ball bearings varies as the cube root of the applied load and that in roller bearings the stress varies as the square root of the applied load. When the diagram of Fig. 12 is plotted in the usual terms, stress versus life, as in Fig. 13, we find that the slopes for ball bearings are 9 and 12, while for roller bearings the slopes are 6 and 6.66. The steeper slope for roller bearings presumably is the result of increased stress at the end of the roller due to the discontinuity of shape which thus constitutes a region of stress concentration. This further confirms the hypothesis that slope is a measure of stress.

Fig. 14 is a diagram of catalog rated load versus catalog rated life in terms of inner race revolutions for several makes of ball bearings of the same size races and the same size and number of balls. Note again the similarity of rated capacity for all bearings, the greatest difference being that one manufacturer rates his bearings on a fatigue slope of 12 in the 1941 catalog. In previous catalogs this manufacturer rated his bearings on a stress fatigue slope of 9 from which we may assume that he has either improved the durability of his bearings or that he has elected to rate his bearings on a new average line within the scatter band. The diagram, Fig. 14, is also interesting in that it does not show a fatigue limit[8] as is usually found in fatigue specimens. According to the catalog ratings, the sloped lines continue to more than a billion inner race revolutions and, since there are several stress cycles per revolution, we do not find a "knee" in these curves up to more than five billion stress cycles.

We seek to determine true stress only as a step in predicting the adequacy or inadequacy of our designs. Any other means that will enable us to predict the performance of our designs will do as well. Ball and roller bearing manufacturers do not consider stress at all in their catalog ratings, but rely entirely upon tabulated load capacities as determined by service experience correlated with laboratory test data on complete bearings. In practice, we are not only unable to calculate or to measure stress, but we do not even know the manner of load applications in service on the majority of machine parts.

The somewhat common belief that we can conduct reliable tests in the laboratory by reproducing the conditions of service is wholly erroneous. By the time the laboratory investigator has provided for all of the conditions that occur in service, he will, in the case of automobile parts, find himself on the road with a complete automobile and, even, then, he will not represent the type of driver who most severely taxes the strength of the machine. Many laboratory tests have been used, and are still being used, by which to grade materials and processes that are now known to have been very costly to the automobile and other industries. Thus, the fiction that a carburized part should have a hard case to resist wear and a tough core to resist breakage arose from laboratory im-



■ Fig. 13 – Characteristic fatigue curve slopes, stress versus life – These are the same curves as shown in Fig. 12 except that load is converted into stress according to Hertz equations

pact tests. In this test, the strength of the part was judged by the number of hammer blows a part would withstand before fracture and since, for example, gear teeth resisted impact fracture in accordance with the physical properties of the core, it seemed logical to specify heat-treatments to bring out the best compromise between the imagined requirements of the case and the core. Being compromises, these heat-treatments were not the best for either region. If, instead of counting the number of hammer blows to produce fracture, the gear tooth had been examined after the first impact, the tooth would have been found bent and, therefore, ruined. Hence it would make no difference how many more blows were required to fracture the tooth. This compromise heat-treatment resulted in reducing the quality of many millions of gears before it was realized that gear teeth fail by fatigue and that fatigue failure, for the usual depth of carburization, always originates at the surface of the case. It then became clear that the heat-treatment should consider the requirements of the carburized case only and that the properties of the core were relatively unimportant since the core serves mainly as a stuffing for the case.

Similarly, gear steels and steels for many other parts have long been selected by false standards that are based

upon laboratory tests, among which are fatigue tests of ideal specimens. For many years, industry has paid premium prices for alloy steels because of their fancied advantages when used in gears and in other parts. Fatigue tests on actual machine parts correlated with service records have shown that there is no detectable difference between the high-priced alloy steels and the low-priced alloy steels when used in many machine elements. This result is probably due to the fact that, as fatigue specimens, machine parts are so far removed from the ideal laboratory fatigue specimen that the latter is misleading as a measure of worth. This is indicated by the relative slopes of the fatigue curves since, as already stated, the slope of fatigue curves for machine parts is always steeper than the slope of fatigue curves for the preferred laboratory specimen by which a material usually is judged.

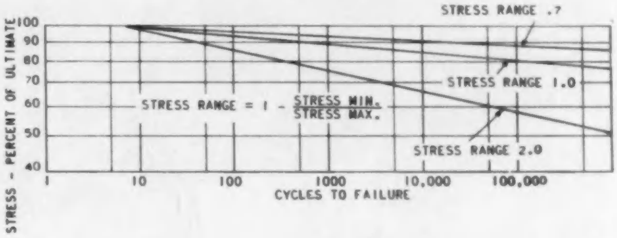
Efforts to improve products by improving surface finish may sometimes have the opposite effect. Highly finished fillets may lead to a false sense of security if they are applied to parts having high internal stresses resulting from manufacturing operations or, as in ground gear teeth, the grinding operations may introduce high surface stresses in tension and thus promote fatigue failures. It is the writer's opinion that, from the standpoint of strength, more harm than good results from the grinding of gears. The surface stresses from grinding are often so great as to produce visible or magnaflux surface cracks but, whether detectable or not, surface tension is frequently very serious. Since fatigue cracks start on the side of the gear tooth that is loaded in tension, the effective stress is the grinding pre-stress plus the working stress. Laboratory fatigue testing of automobile or other light-weight, high-output machine parts, as well as other laboratory tests such as on fuels, oils, tire wear, and so on, must be correlated definitely with service data on the part in question before the results can be accepted. This requires that, for fatigue, tests must be devised that will agree with failures that occur in normal service as to the location of points of fracture and the character of the fractures whether or not the test procedures agree with preconceived notions of service loading.

The slope of the finite life portions of the S-N diagram has been discussed from the standpoint of fatigue tests at constant stress range. Most of the test data presented hereto have been taken from specimens in which the stress was completely reversed. However, many machine parts

■ Fig. 14 – Catalog rated load versus catalog rated life of four makes of ball bearings, all of same size, to show differences in load ratings of the different makers

are otherwise stressed as, for example, valve springs which are stressed through a relatively narrow range in one direction only. Valve springs are preloaded to approximately 25,000 psi stress which is increased to approximately 90,000 psi when the valve is fully open. Gear teeth usually are loaded from zero stress to a maximum stress in one direction only. Axle shaft stresses are somewhat more complex being completely reversed in bending during each revolution due to the weight of the car while transmitting torque in one direction only.

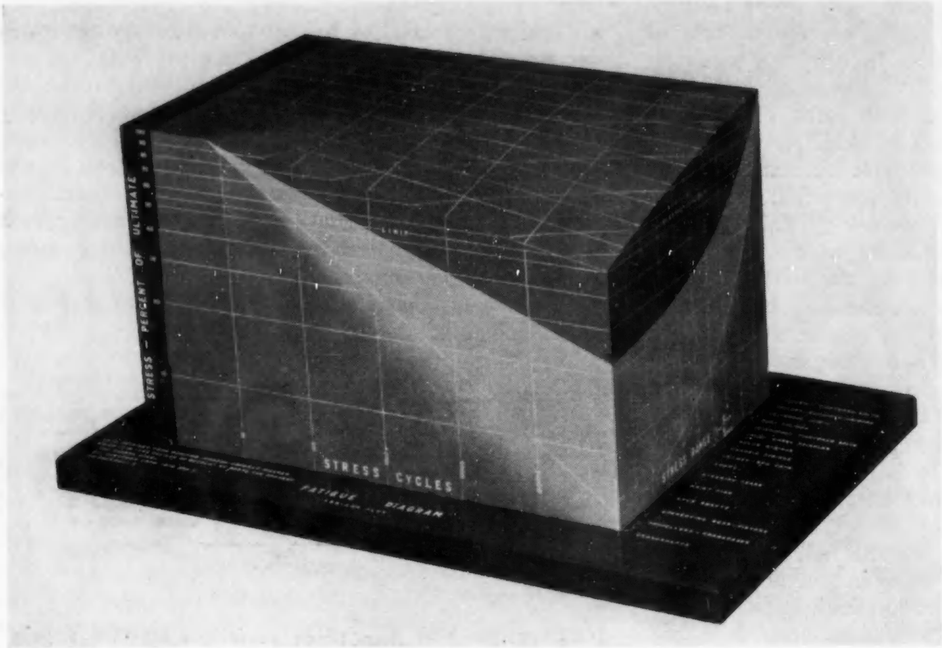
Many experiments have been conducted to determine



■ Fig. 15 – Effect of stress range on fatigue life – The slope of fatigue curves varies with stress range as well as with stress concentration

the effect of varying the stress range, but again interest lay in the stress at the fatigue limit and little data are available on the change of slope with stress range. However, since the stress at the fatigue limit increases as the stress range decreases as has been amply demonstrated, it follows that the slope must decrease (become flatter) as the stress range decreases. Moore and Kommers[2] present, somewhat apologetically, a modified Goodman diagram from which the fatigue slope for ideal specimens may be constructed for any stress range. Fig. 15 shows a re-plot of this modified Goodman diagram for three stress ranges. The upper curve represents a small stress range similar to that of automobile valve springs; the second curve represents a stress range of from zero to maximum as for gears; and the lowest curve represents complete reversal of stress as may occur in a crankshaft. The slopes of these curves are respectively 80, 48, and 17. We thus see that the slope of the fatigue curve varies with stress range as well as with stress concentration and, therefore, the hypothesis that the slope of the fatigue curve is a function of stress is no longer tenable. If, however, we state that the slope of the fatigue curve is a function of effective stress, the hypothesis will apply for any stress range.

Fig. 16 is a three-dimensional diagram on logarithmic coordinates of the modified Goodman diagram by Moore and Kommers. The vertical scale of this diagram represents stress, the horizontal scale on the forward side represents stress cycles, and the horizontal depth scale represents stress range. The forward face is the ordinary S-N diagram for complete reversal of stress; the back face is an S-N diagram for a stress range of zero; and a section at the middle of the stress-range scale would be an S-N diagram for a stress range of zero stress to maximum. The numerical values of the stress-range scale have been arbitrarily selected so that 2 represents complete reversal of stress; 1 represents stress from zero to maximum; and zero represents no change in stress. The equation $R = 1 - \frac{\text{minimum stress}}{\text{maximum stress}}$ has, therefore, been used by

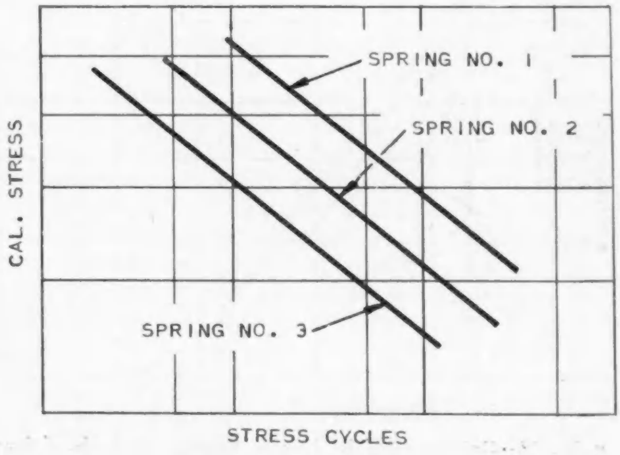


■ Fig. 16 – Three-dimensional fatigue diagram showing the effect of stress range on the slope of the fatigue curve

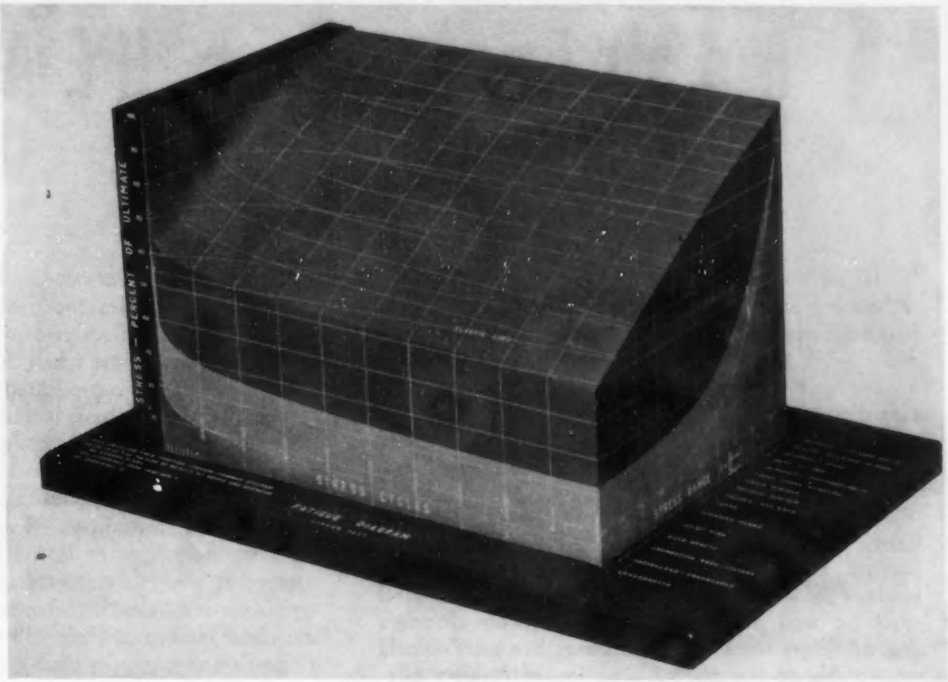
which to locate points of intermediate stress range. On the forward face of this diagram are drawn two straight lines diverging from a point at the upper left. These two lines represent approximately the best slope (upper) and the poorest (lower) that have been obtained from available fatigue tests on machine parts at complete stress reversal. These slopes are such that, at one million stress reversals, the poorest specimens (slope 4) are stressed, by hypothesis, between 7 and 8% of the tensile strength of the material, the best specimens (slope 8) are stressed, by hypothesis, to about 23% of the tensile strength, whereas the slope of the Moore and Kommers' diagram (slope 17) is stressed to 50% of the tensile strength. These conclusions are based upon the further hypothesis, previously stated, that fatigue diagrams are straight lines, on logarithmic coordinates, to a point on the tensile strength line at some considerable number of stress cycles. Since the data on machine elements indicating the slopes of their fatigue curves are meager, the diagram should be considered as qualitative only. The stress-range (depth) face also shows lines that are even less supported by reliable evidence. They also represent the best and the poorest of fatigue tests on machine parts and are based on one point each at stress range 2, at stress range 1 and converging to a point at stress range zero and ultimate stress. The diagram is presented here in the hope that other experimenters will come forward with data to prove or disprove the hypothesis upon which it is based.

Fig. 17 is the same diagram as Fig. 16 except that it is plotted on linear coordinates better to present dimensional values, particularly to those who are not accustomed to the use of logarithmic charts.

Since we have no reliable means for determining stress and since fatigue tests on laboratory specimens cannot be used for evaluating the strength of machine parts, we have no recourse but to continue fatigue tests on machine parts in our industrial laboratories. There is, however, much that we can do to improve our technique in setting up the conditions of tests and in interpreting the test data. The methods now used for coordinating laboratory tests with service are too haphazard to be completely reliable. Service failures must, obviously, be infrequent and when true fatigue failure does occur, it is the result of harder-than-usual service combined with a point lying on the lower fringe of the fatigue scatter band. Since failures must be infrequent, it is highly important that failed parts be examined by competent observers in order that the true cause of the trouble may be determined. Clear evidence of fatigue failure does not prove that the failed part was primarily responsible. A bolt may fatigue because it was not properly tightened during assembly; a gear may fatigue due to improper support or to a failed bearing; a crankshaft may fatigue due to inadequate or maladjusted vibration damper; and so on without end. It sometimes happens, therefore, that immediate corrections are made to the wrong part and recognition of the true trouble is sometimes greatly delayed. Laboratory fatigue tests on machine parts must not only duplicate service failure as to location of fracture, but they must, in many cases, produce failure in approximately the same number of stress cycles if accurate life comparisons are to be made. This requires that

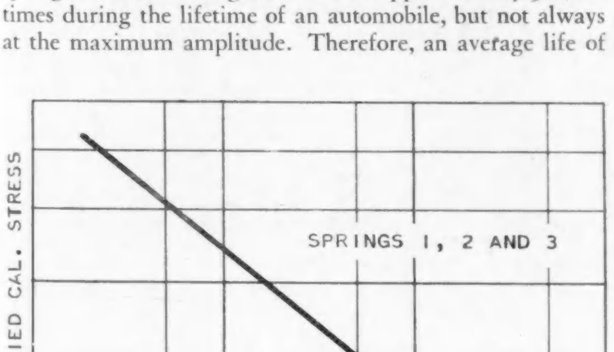


■ Fig. 18 – Inadequate formula



■ Fig. 17 – Same as Fig. 16 except that linear coordinates are used

we distinguish between normal operation stress and the relatively infrequent overloads that caused the failure. Rear-axle automobile gears are, at worst, stressed at low-gear torque one cycle out of every one thousand cycles. The lifetime requirements of such gears, therefore, is 100,000 cycles at low-gear torque equal to approximately 30 miles of travel. Due to the scatter of test points this is approximately 250,000 cycles measured on the average fatigue curve. Chassis springs normally operate through a small stress range, but they must be designed to withstand a total of high stress cycles equal to the number of bumps that will be experienced by the hardest driver on the worst road. This is a relatively small number of stress cycles but the problem is aggravated by the fact that such springs are subjected to severe corrosion and to surface damage by stones. Hence experience requires an average lifetime of 100,000 cycles at maximum amplitude. Clutch springs in town driving are deflected approximately 500,000 times during the lifetime of an automobile, but not always at the maximum amplitude. Therefore, an average life of 500,000 cycles at full amplitude is a minimum requirement.



■ Fig. 19 – Modified formula

Fatigue data from machine parts can be studied profitably for the purpose of constructing empirical formulas by which load capacity eventually may be accurately calculated. Let us assume that we have accumulated fatigue data on one hundred coil springs each of three quite different sizes. We calculate the stress and plot these points on a logarithmic chart, as shown in Fig. 18, and we find that all springs of one size lie within a definite scatter band; all test points of the second size spring also lie within a definite scatter band having the same mean slope as the first group but somewhat lower on the chart; and the third set of springs plot like the other two but at still another level. From this plot we may conclude immediately that the formula used for calculating stress is in error since, by its use, we were unable to predict relative load capacity. If we now introduce into the formula a suitable size factor such as to bring the three curves and their scatter bands into coincidence, as shown in Fig. 19, we will have improved the accuracy of our calculations. This is the method that has been followed in arriving at the load capacities of ball and roller bearings and for automobile gears. By this procedure we may construct accurate load-capacity formulas for any machine part no matter how complex the stress pattern may be. A large number of fatigue test points are required, but this need not be a serious obstacle. Fatigue tests have been made for many years by many laboratories and there are, therefore, now sufficient data, if they can be assembled, to construct formulas for a considerable variety of machine parts, with more data being accumulated every year. The stress calculated by such formulas will not be true stress but this is unimportant because we will not have reliable means by which to determine load capacity. If, at some future time, it can be shown that the hypothetical fatigue curve, Fig. 6, or some equivalent system, applies to structural materials, it will become possible to determine true stress. It is suggested that this latter phase of the problem can best be entrusted to the laboratories of our technical schools.

Bearings for HEAVY-DUTY

THE author points out that certain specialized bearing materials, such as the cadmium alloys, together with mixtures of copper and lead, will do more work than tin-base babbitt, but they insist on doing it in their own way. His paper consists essentially of a technical explanation of the peculiarities and characteristics of the higher-capacity bearings made from these specialized materials.

Data are presented to show that increased bearing mileage is obtained by the use of thinner linings. On the other hand, it is emphasized that engines in which thin babbitt bearings are used must be well protected by efficient air cleaners and oil filters since these bearings are particularly susceptible to the abrasive action of foreign particles in the lubricating oil.

In a discussion of replacement bearings, the unfavorable effects of the following conditions are emphasized: worn and out-of-round crankpins and journals; bowed crankshaft retained in line by force; warped crankcase; out-of-round connecting-rod bores and main bearing saddle bores; dirt within the engine; bearing caps misplaced sidewise; and improper clearance.

In a section on lubrication, Mr. Willi shows that the addition of a centrally located internal annular oil groove will improve the load capacity of bearings with a high *L/D* ratio.

Considerable space is devoted to a discussion of copper-lead bearings which, the author believes, will come into increasing use. This section is concluded by 15 rules for the installation of copper-lead bearings.

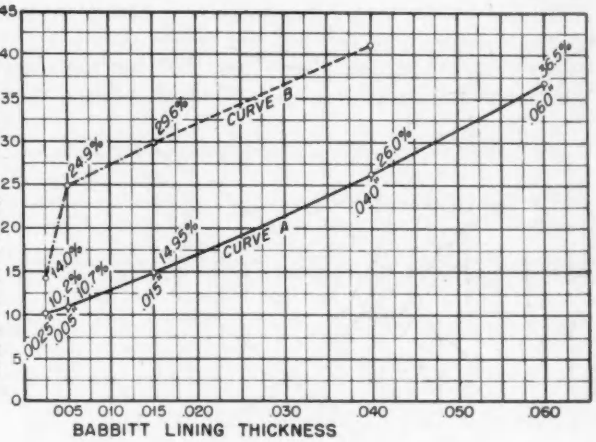
TECHNICAL literature is replete with papers, articles, and textbooks dealing with sleeve-type bearings. Most bearings in an engine are sleeve bearings as distinguished from ball, roller, or needle bearings. Much of this literature is particularly interesting because of the wide differences in opinions expressed or the wide differences in the interpretation of test results.

The profusion of literature probably indicates the relative importance of the subject because bearings of some sort must be used wherever shafts move and wheels turn.

The main and connecting-rod bearings in an internal-combustion engine have a difficult job to do. They must go about their work quietly, unobtrusively, and efficiently, taking the rap of tremendous power thrusts and the effects of high speed with a minimum of distress and complaint so that the power-developing elements can produce the energy that moves the vehicle and its load quickly and economically.

There was a time when the engine bearing situation was not particularly complicated because the accepted bearing metal was a tin-base babbitt of the general composition 3.5% copper, 7.5% antimony, and 89% tin. It was temporarily adequate for the work that it had to do, but eventually the need for specialized higher-capacity bearings and bearing materials arose. They were duly forthcoming, most prominently perhaps in the form of certain cadmium alloys together with mixtures of copper and lead. These materials do not have the same lovable characteristics as do the tin-base babbitts. They will do much more work but they insist on doing it in their own way, and it is necessary to understand thoroughly the peculiarities and characteristics involved. With a possible shortage in tin and an existing shortage of cadmium in the picture, the status of the copper-lead bearing becomes increasingly important.

Tin-base babbitt possesses extremely valuable properties for bearing construction, but some of its physical properties are relatively low – compression, tension and hardness – and these properties drop off drastically as temperature increases.[1] These weaknesses have been minimized and bearing capacity increased by chemical adhesion of the babbitt to the back structure, improved and controlled babbitt structure, and by steadily decreasing the babbitt thickness. In one of the Society of Automotive Engineers' papers of 1932,[2] bearings with a babbitt thickness of 0.005 in. are discussed at considerable length and the conclusion is drawn that a lining of this thinness will add to the mileage expectancy of a bearing, but manufacturing



■ Fig. 1 – Relationship between babbitt thickness and area of babbitt failure

Automotive Engines

by ALBERT B. WILLI
Chief Engineer, Federal-Mogul Corp.

methods are not yet (1932) developed to a point where this is practical.

At this writing, steel-back babbitt-lined bearings with the lining as thin as 0.0025 in. are being produced and used in several engine models.

Characteristics of Babbitt-Lined Bearings

That increased mileage is obtainable by the use of thinner linings is illustrated by the curves in Fig. 1 which show a certain relationship between babbitt thickness and area of babbitt failure. Curve *A* is plotted from the results of 100-hr tests in a special bearing test machine[3] using steel-backed, "precision-insert" connecting-rod bearings lined with "genuine" babbitt (3.5% copper, 7.5% antimony, and 89% tin) in thicknesses of 0.0025, 0.005, 0.015, 0.040, and 0.060 in.

"Area of failure" is defined as those portions of the babbitt surface showing distress in the form of cracks of any pattern and broken out sections of babbitt. In the bearings tested, these areas were carefully traced, measured with a planimeter and recorded as a percentage of the total developed bearing area.

Curve *B* is plotted from engine test data. The curve does not possess the uniformity of Curve *A*, but the value of the reduced babbitt thickness is evident. The dotted extension of Curve *B* is not supported by additional corroborating tests.

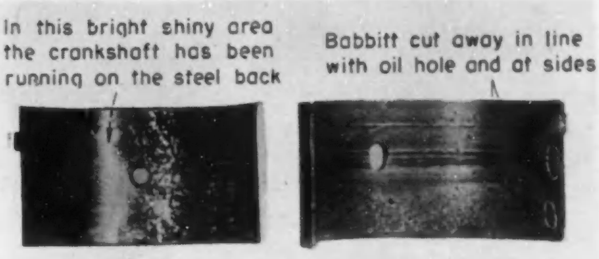
Lest we become overly enthusiastic regarding the microscopically thin babbitt, let us examine the possibilities for trouble.

One of the important properties of tin-base babbitt is its ability to imbed and absorb a reasonable number of foreign particles without damage to the bearing itself. It is not uncommon to find foreign particles (cast-iron chips, road dirt, and so on) in the oil stream which are as much as 0.005 in. thick. When these start to work on a babbitt lining 0.0025 in. thick, both the bearing and shaft must suffer.

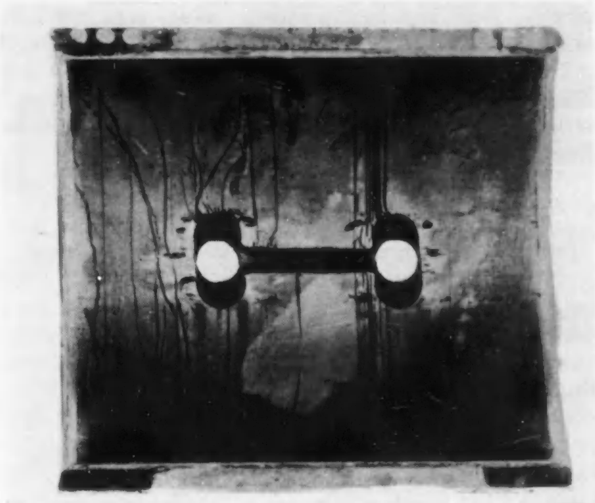
The bearings in Fig. 2 clearly show how the very thin babbitt was worn and cut away by excessive amounts of foreign particles which contaminated the lubricating oil.

The bearing in Fig. 3 had a normal babbitt thickness. Its point of interest is the scoring pattern and the certainty that foreign particles of the size which cut this pattern would be disastrous had the babbitt been extremely thin. The bearing in Fig. 4 also had a normal babbitt thickness and is included to illustrate another pattern of a bearing fouled by dirt.

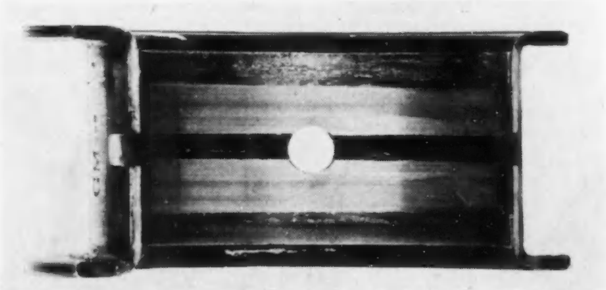
It is imperative that engines in which thin babbitt bear-



■ Fig. 2 – Bearings with extremely thin babbitt linings worn and cut away by excessive amounts of foreign particles in the lubricating oil



■ Fig. 3 – Diesel engine main bearing scored by large foreign particles



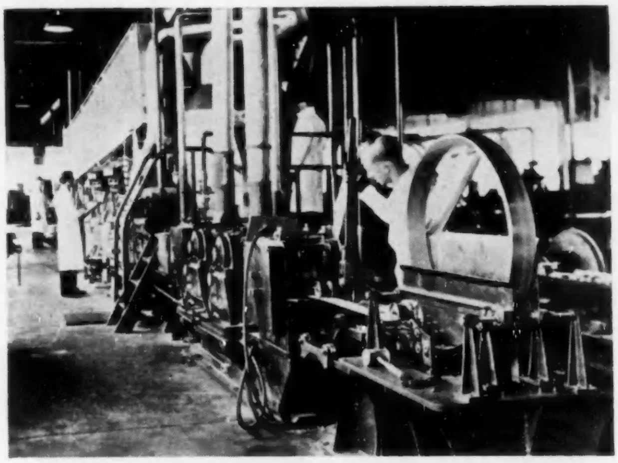
■ Fig. 4 – Passenger car main bearing with heavy dirt imbedments outside the limits of the distributing grooves

[This paper was presented at the West Coast Regional Transportation and Maintenance Meeting of the Society, San Francisco, Calif., Nov. 5, 1941.]

[1] See SAE Transactions, December, 1939, p. 514: "Engine Bearings from Design to Maintenance," by A. B. Willi.

[2] See "The Manufacturer's Viewpoint on Engine Bearings," by D. E. Anderson, presented at the Annual Meeting of the Society, Jan. 26, 1932.

[3] See Symposium on Lubricants, American Society for Testing Materials, March 3, 1937: "Automotive Bearings – Effect of Design and Composition on Lubrication," by Arthur F. Underwood.



■ Fig. 5 – Mill for applying babbitt surface to a continuous steel strip

ings are used must be well protected by efficient air cleaners and oil filters.

When a bearing lined with a normal babbitt thickness (0.015 in. and upwards) "burns out," the extra clearance between shaft and back structure causes sufficient noise and disturbance to warn the operator that something is radically wrong. If a bearing having a 0.0025-in. babbitt lining burns out, the noise and disturbance will certainly be less and the operator may or may not realize immediately that he has trouble on his hands.

■ Availability of Undersize Bearings

In operation and maintenance, availability is important – availability of the vehicle and availability of parts when replacements are necessary. The mileage obtained from a number of the originally installed critical parts in an engine such as spark plugs, valves, pistons, piston pins and rings, connecting-rod and main bearings, and so on, is only a moderate percentage of the total mileage obtainable from the major structural and heavy members. Therefore, in the total life of an engine, these critical parts will have been replaced a number of times.

Crankshafts are not often subject to breakage, but the crankpins and journals are subject to wear and must be reground. Reground crankshafts and undersize bearings are probably used for many more miles per engine than standard shafts and bearings.

It is standard practice that undersize bearings be made with the same back thickness as standards and with the variation for shaft undersizes obtained by increasing the lining thickness. The reasons for this involve both economy and availability. Undersize bearings are in demand to be used with shafts from 0.001 to 0.090 in. undersize. Sometimes the undersizes are in increments of 0.010 in. under standard but, in many cases, the undersize is an odd dimension.

With the highly specialized equipment used in the bearing industry today, such as the mill shown in Fig. 5 used for applying a babbitt surface to continuous steel strip, it would be extremely costly to maintain a standard babbitt thickness and obtain the undersize variable in the back thickness. Existing practice permits standard and undersize bearings to be run through production en masse up to the final bore-sizing operation. To maintain standard linings would require complete separation of standard sizes and comparatively small quantities of a number of different undersizes right from the start. The effect on cost of undersize bearing production needs no elaboration.

To provide material availability without excessive inventory, many dealers and large operators carry only 0.060 or 0.090-in. undersize bearings in stock which are rebored on the premises to any required undersize. Sometimes these bearings must be finished up as standards.

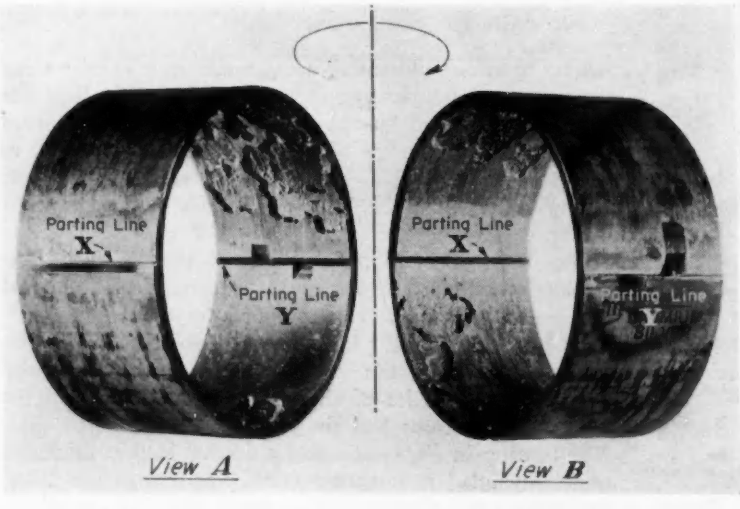
It is thus apparent that, while full advantage can be taken of the thinner linings in original-equipment bearings, their application for maintenance and replacement offers some interesting problems.

■ Load Capacity of Replacement Bearings

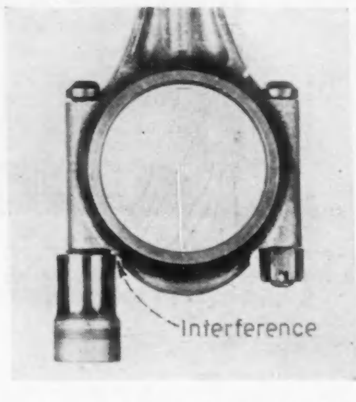
It is rather common to be faced with the question:

"The original bearings in this engine were good for 150,000 miles. Similar bearings installed as replacements only lasted 75,000 miles. WHY?"

The degree of renewal or extent to which an engine's youthfulness is restored during a rebuilding operation or at the time new parts are installed varies widely and we



■ Fig. 6 – Pattern of premature failure in a connecting rod bearing caused by a misplaced cap



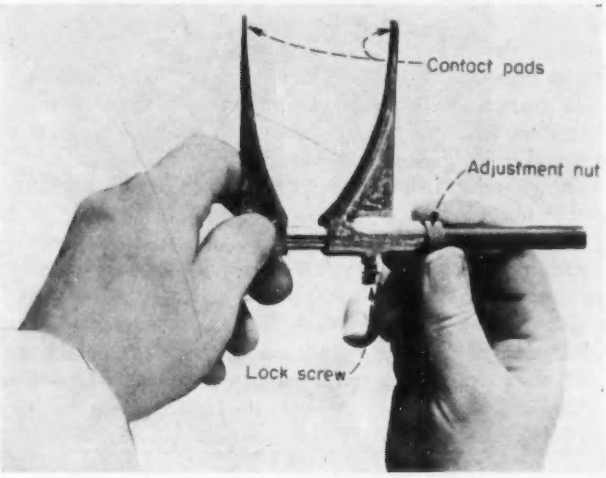
■ Fig. 7 – Corner of oversize nut socket pushes cap sidewise

must differentiate between a thorough rebuild and a routine repair or replacement job. It is obvious that, in a routine replacement job performed with a minimum amount of engine dismantling, there will probably exist such unfavorable conditions as worn and out-of-round crankpins and journals, a bowed crankshaft retained in line by force – imposing a heavy and false load on the main bearings – a warped crankcase with the main bearing saddle bores out-of-line which also imposes false loads on the mains, out-of-round connecting rod bores and main bearing saddle bores, excessive amounts of dirt within the engine picked up due to carelessness, bearing caps misplaced sidewise, and improper oil clearance.

I want to enlarge a little on the misplaced cap item at this point and show Fig. 6, the pattern of failure in a connecting-rod bearing caused by a cap which was misplaced sidewise.

In View *A* the lining is a total wreck above parting line *Y* and sound below it. View *B* was obtained by moving the bearing 90 deg to show the opposite surface where the conditions are reversed. The lining is sound just above parting line *X* and in very bad shape directly below it.

The cause of this sort of misalignment often can be traced to the socket wrench used (see Fig. 7), unless positive cap doweling is provided.



■ Fig. 8 – Crankshaft caliper

The diameter of the nut socket is too great. A corner bears against the cap and, if the operator exerts much pressure endwise, the cap will be pushed sidewise when the first nut is tightened.

The oil-clearance item also deserves some special mention. Bearings must have proper clearance over the shaft and, if main bearings are to be installed without removing the shaft, the determination of the journal size has been largely a matter of guesswork. This guesswork can be eliminated largely by the use of a recently developed "crankshaft caliper" (Fig. 8).

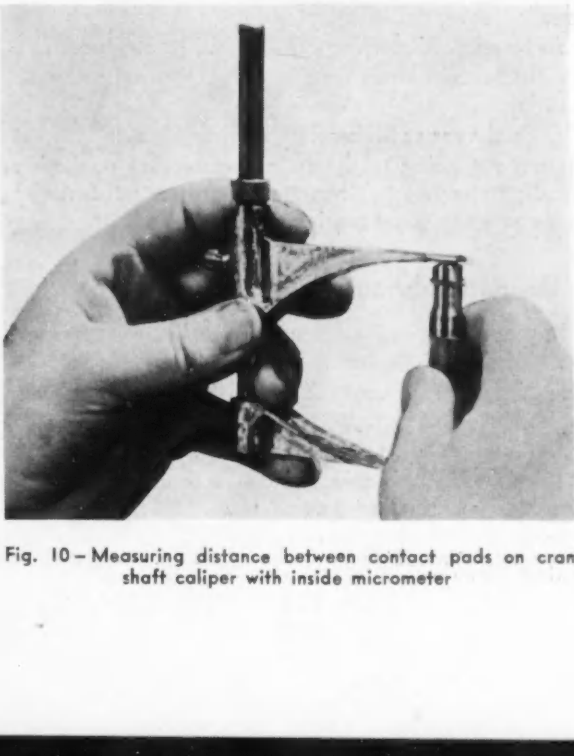
This caliper is suitable for use in all engines in which the main bearings can be removed and replaced without disassembling the crankshaft. With the old bearings rolled out, it is possible to reach in and readily obtain the journal size, as shown by Fig. 9. The actual dimension is finally taken with an inside micrometer as shown in Fig. 10.

Detonation (spark or fuel knock) is a trouble breeder for bearings. The effect of detonation is heavily to increase

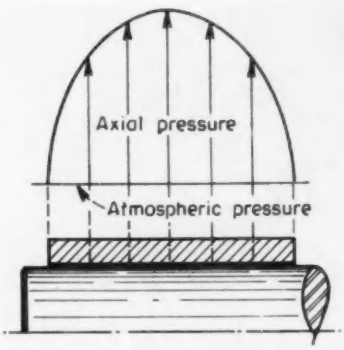
■ Fig. 9 – Obtaining size of crankshaft journal

the loads on the main and rod bearings. If new rod bearings are installed as part of a complete rebuild, carbon deposits in the combustion chamber and on the piston heads naturally will be removed. If, however, the rod bearings are installed independently of a complete rebuild, the carbon probably will remain to induce detonation. Even in a rebuilt engine, there may be a considerable tendency toward this ailment if the radiator has not been cleaned thoroughly to insure free flow of cooling water, and impeding deposits of lime, scale, and so on, removed from the engine water jackets. These things, along with hose connection renewals, are not always a part of a rebuild.

Low oil pressure as an indication of reduced oil flow through the lubricating system, will reduce bearing mileage. The most obvious cause of low oil pressure is excessive wear and clearance at the main and rod bearings. These are also the easiest points at which to make a correction,

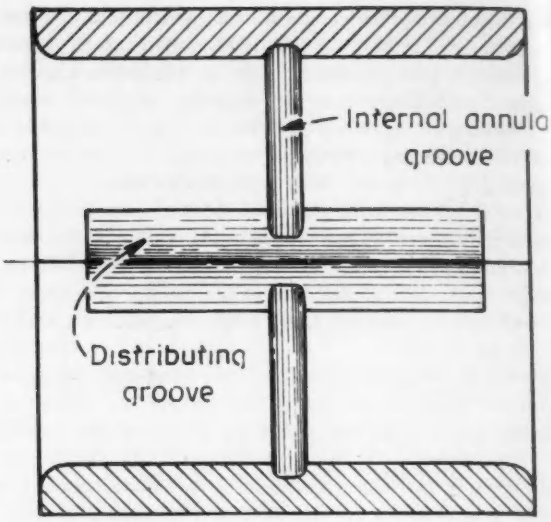


■ Fig. 10 – Measuring distance between contact pads on crankshaft caliper with inside micrometer



■ Fig. 11 (left) – Diagram showing longitudinal pressure distribution in a bearing oil film

■ Fig. 12 (right) – For bearings lubricated through a drilled shaft, an internal annular groove to feed the distributing grooves is often required



but low oil pressure may be caused by excessive camshaft bearing wear, a weakened oil pressure relief spring which permits too much oil to be bypassed without traveling through the system, and also by worn oil-pump gears and housings. These latter items are often disregarded even in a rebuild.

The gliding surface of the piston skirt must be square with the connecting-rod bearing bore. If out-of-squareness exists, false loads of high magnitude are placed on the connecting-rod bearings – also on the piston skirts and cylinder walls. It is not difficult to maintain proper squareness limits on these parts in the engine plant but, in many rebuilding operations, the connecting rod is aligned to square up the piston by twisting or bending it with a notched bar. I think that it is safe to say that heavy rods seldom remain aligned after this twisting and bending operation. The steel is not permanently set and the rod soon returns to its warped condition.

All of this goes to show that main and connecting-rod bearings (and many other engine parts for that matter) installed as replacements in engines which have seen hard service will have a much more difficult job to do than did the original parts. It is therefore unreasonable and impossible to expect the second or third set of parts to show the same mileage as did the first – assuming that the replacements are of equal mechanical quality and capacity. In many cases it is possible to obtain the original mileage or possibly to exceed it, even in spite of the described handicaps, if bearings of increased load-carrying capacity are installed as replacements.

Increased load-carrying capacity can be obtained:

a. In a moderate measure by manufacturing methods.[4]

b. By improved lubrication.

c. By the use of higher-duty bearing metals.

I am not going to discuss manufacturing methods and details; Reference (4) handles the matter satisfactorily in so far as white metal bearings are concerned.

■ Bearing Lubrication

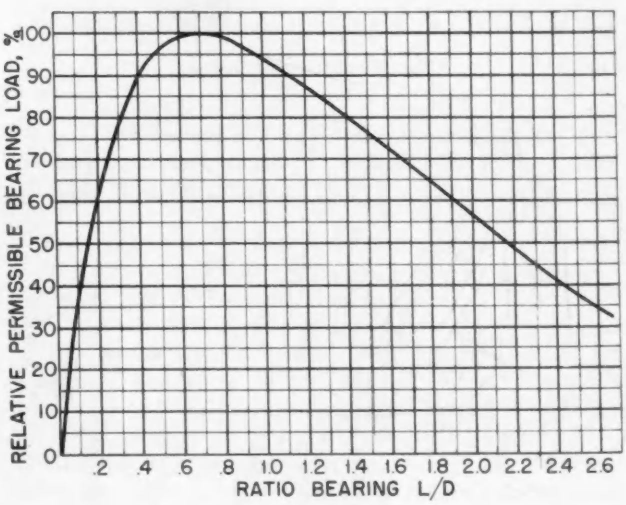
Improved lubrication might involve:

1. A more suitable oil specification.
2. A greater volume of oil delivered to and passed through the bearings.
3. The installation of an effective oil cooler.
4. Directed lubrication within the bearing by means of *correct* grooving rather than haphazard lubrication without any grooving or perhaps an utterly incorrect grooving system.

The purpose of the delivering of and passing through, a greater volume of oil is to reduce the bearing temperature, a function of oil which is at least as important as its function to lubricate. It is not the intent to suggest particularly that new oil pumps of greater capacity than the original design be installed as a part of an engine rebuilding or overhaul program. The impracticability of obtaining such pumps is obvious. It *is* important, however, that the oil-pump delivery and oil pressure in a rebuilt engine be restored to its original condition.

Oil grooving in a bearing seems to be a rather controversial item.

The familiar diagram showing longitudinal pressure distribution in a bearing oil film (Fig. 11) is often a deterrent to making valuable use of an annular oil groove within a bearing. In certain bearings wherein pressure lubrication is provided through a drilled shaft – as in the case of a connecting-rod bearing – an internal annular

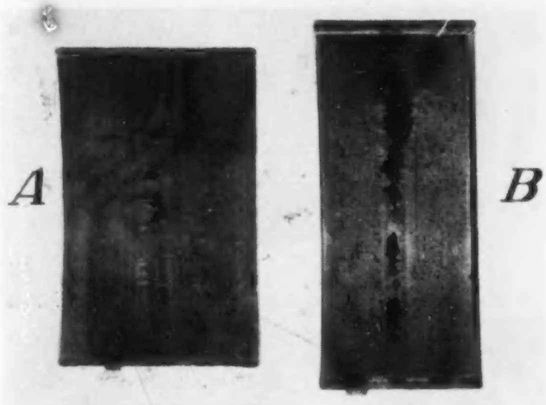


■ Fig. 13 – Relative permissible bearing load plotted against $\frac{L \text{ (Length)}}{D \text{ (Diameter)}}$ ratio
(from *Automobiltechnische Zeitschrift*, September, 1932)

[4] See SAE Transactions, December, 1939, pp. 520-521: "Engine Bearings from Design to Maintenance," by A. B. Willi.

groove as shown in Fig. 12 located in registry with the shaft oilway is sometimes very effective in improving bearing life. Oil distributing grooves in engine bearings are fairly common but, if the shaft speed is high and the bearing is rather long (approximately in excess of 1 to 1½ times the diameter), the period of registration between the shaft oilway and the distributing groove in the bearing may be insufficient to fill the grooves completely and keep them filled. If oil is lacking in quantity, obviously the proper film cannot be formed and the conditions indicated by the pressure diagram cannot be maintained.

The internal annular feeder groove in Fig. 12 insures that the distributing grooves will always be fully supplied with oil and a complete film obtained. In any given bearing, the addition of such a groove reduces the projected bearing area and increases the theoretical unit pressure, but a correctly lubricated bearing will function longer with higher loads than an improperly lubricated one will with considerably lower loads.



▪ Fig. 14 – Engine bearings with areas of failure in line with the travel of the crankshaft oilways

An internal annular oil groove is not always a benefit, but a clue as to whether it might or might not be helpful is found in Fig. 13, a curve showing "relative permissible bearing load plotted against the $\frac{L \text{ (Length)}}{D \text{ (Diameter)}}$ ratio."

The L/D ratio for maximum efficiency is shown as 0.7 to 1.0. Load-carrying capacity drops off very rapidly as L is reduced and considerably less rapidly as L is increased.

A bearing divided by an internal annular groove is in fact two bearings so, using this curve as a basis, it is apparent that an internal annular groove applied to a bearing having a lesser L/D ratio than 0.8 to 1 will seriously reduce its load-carrying capacity. (If the ratio is 0.8 to 1, each longitudinal half section will have an L/D ratio of 0.4 to 1 with the load-carrying capacity reduced to 90%.) As the bearing increases in length, the value of the internal annular groove is apparent, as for example:

A given bearing with an L/D ratio of 1.5:1 will carry a relative load of approximately 75%. If a centrally located internal groove is added, we have in fact two bearings, each with an L/D ratio of 0.75:1 with a relative load capacity of practically 100%.

Aside from these considerations, an internal annular groove is often a necessity if premature failure of the bearing surface in line with the travel of the crankshaft oilway is to be avoided. In Fig. 14, *A* is the upper half of a connecting-rod bearing and *B* is the lower half of a main bearing. In both cases the oilway broke through in the high pressure area of crankpin and journal.

A somewhat similar condition is shown in Fig. 15.

The partial oil groove was probably used in the original bearing at *A* with the idea of reducing end leakage and oil throw-off, but premature breaking up of the babbitt at the trailing edge of the groove spoiled the idea. Correction was made by completing the groove in a narrower width as shown at *B*.

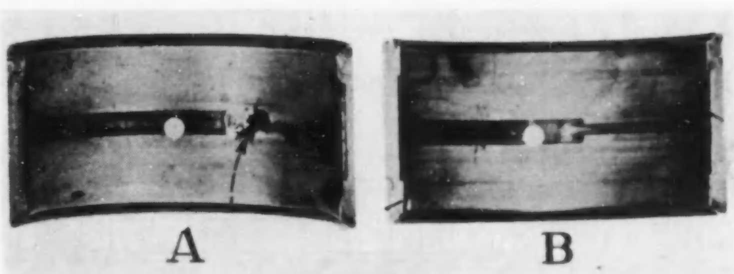
I have devoted considerable time to a discussion of oil grooving for the reason that correctly applied grooving will improve the performance of any bearing assuming, of course, that proper grooves were not originally provided. Conversely, improper grooving will guarantee an unsuccessful bearing. It is comparatively easy to add oil grooves to a stock bearing, and I have seen quite a few hammer-and-chisel grooving jobs of rather weird patterns which insured early failure although their purpose was otherwise.

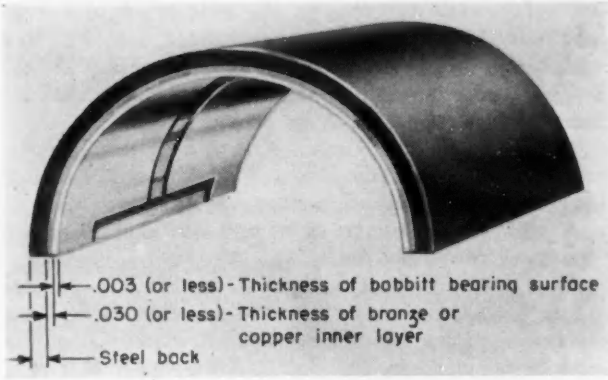
▪ Capacity for Replacement Bearings

It is probably easy enough to recommend that, for engine rebuilding and overhaul jobs, bearings of increased load capacity be installed but, under existing conditions of priority controls and the possibility of these conditions becoming increasingly difficult, the accomplishment of such a plan temporarily may not be easy.

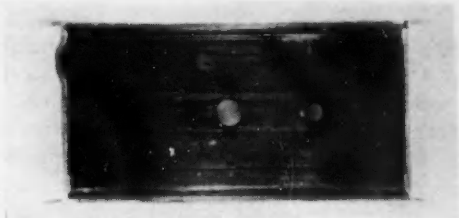
Considering what we may call a conventional tin-base babbitt bearing having a lining thickness of 0.035 in., the first step in obtaining a bearing of increased capacity is to reduce drastically the babbitt thickness to 0.0025 in. or thereabouts. The advantages and disadvantages of this type of bearing were described earlier in my remarks and, in this category, I think it is fair to include certain "multiple-layer" bearings as illustrated more or less diagrammatically in Fig. 16 because they depend primarily on the thinness of the babbitt for increased capacity ratings. As previously pointed out, the availability of bearings of this

▪ Fig. 15 – Babbitt broken out at the trailing edge of partial oil groove in bearing *A* – Failure in this region corrected by completing the groove in a narrower width as shown in bearing *B*

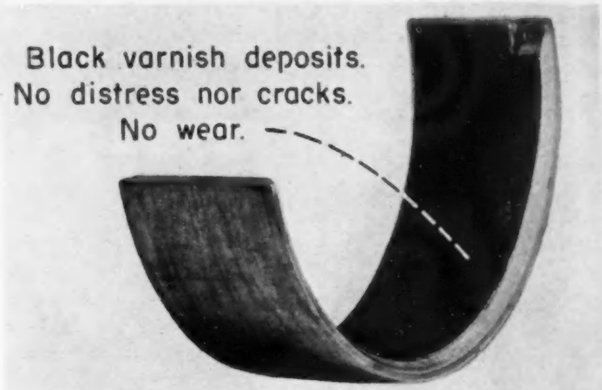




■ Fig. 16 – Bearing constructed with a steel back, an inner layer of bronze or copper, and a very thin babbitt lining



■ Fig. 17 – Copper-lead bearing sent in for an explanation as to why it failed



■ Fig. 18 – Copper-lead bearing diagnosed as a failure

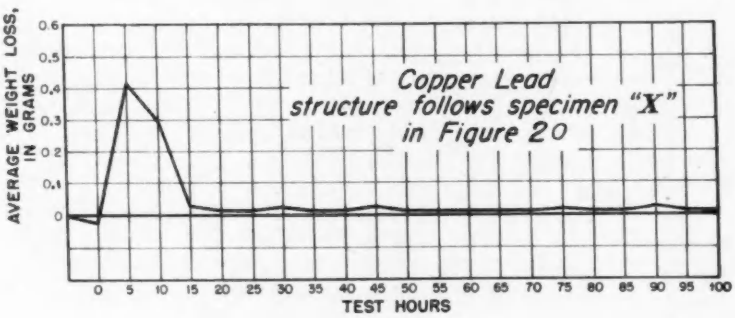
character for overhaul and replacement purposes particularly in connection with crankshafts of odd undersizes is very questionable.

The standard plan used by many operators is to install cadmium-silver-copper bearings in rebuilt or overhauled engines where the original bearings were of babbitt. The greatly increased load-carrying capacity of this material insures mileage performance which is at least the equivalent of the original bearings and often it is considerably greater. This type of bearing has been readily available in any desired undersize.

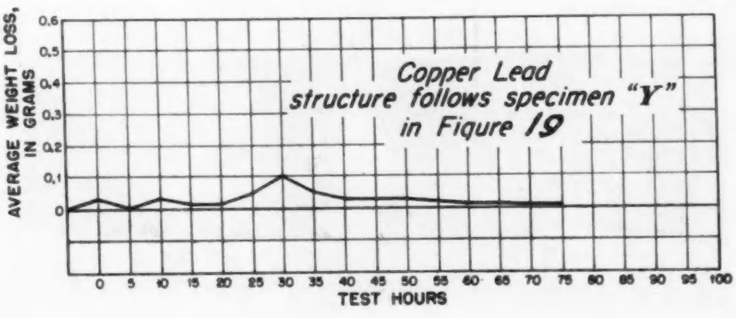
Just how long the cadmium-silver-copper bearings will be available, however, is difficult to predict as cadmium is one of the highly strategic metals.

Next we must consider copper-lead bearings and, in discussing them, I want to cover a few highspots. A great many engines carry this type of bearing as original equipment, but their performance has been spotty. In some operations, they have been marvelously successful; in others, they have been very unsatisfactory. I just completed a survey of the record of performance of copper-lead bearings in a certain engine model across the country, and the results were thoroughly confusing.

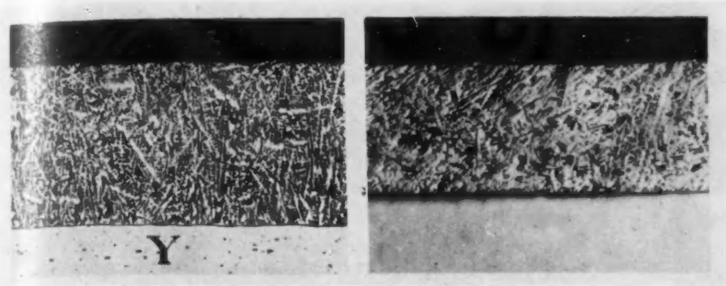
At some points, past experience with copper-leads had been so unsatisfactory that they were removed from new



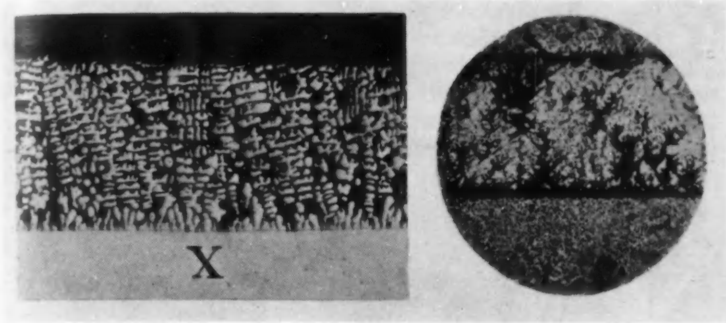
■ Fig. 22 – Engine test to determine rate of copper-lead bearing weight loss due to corrosion and wear



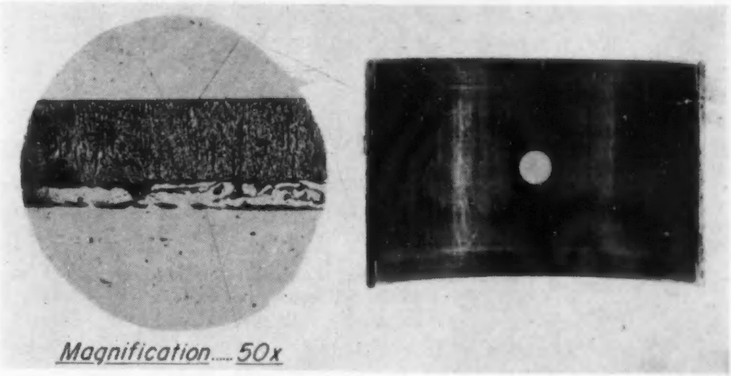
■ Fig. 23 – Engine test to determine rate of copper-lead bearing weight loss due to corrosion and wear



■ Fig. 19 – Copper-lead structures usually found in aircraft-engine bearings – magnification, 50X



■ Fig. 20 – Copper-lead structures common in automotive engine bearings – magnification, 50X



■ Fig. 21 – Copper-lead structure and associated natural surface condition in some automotive engine bearings

engines before the engine was ever put in service and cadmium-silver-coppers installed. At other points, the copper-leads were eminently satisfactory.

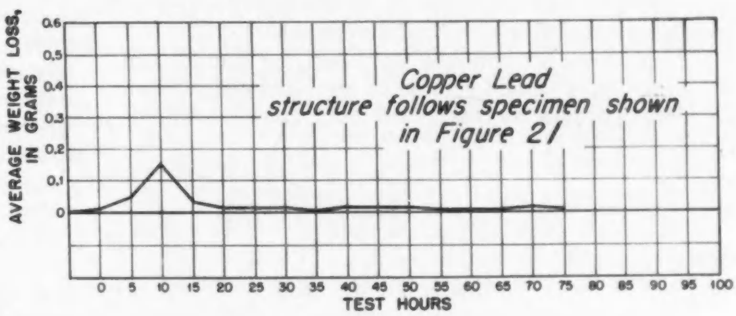
■ Copper-Lead Bearings

I look for an increasing use of copper-lead bearings, not from choice perhaps, but because of necessity. There is plenty of copper and lead in the country, whereas cadmium is a byproduct of zinc manufacture and tin must be imported. Copper-lead bearings are unquestionably sensitive and temperamental, but that simply means that their field of usefulness from an engineering and mechanical standpoint must be known and recognized and that they be applied within the conditions of their field.

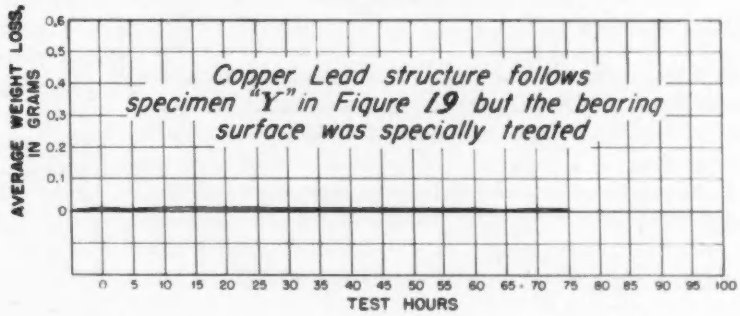
A soft-yielding babbitt bearing can be induced to live for a time with some rather bad engine conditions, but a copper-lead bearing immediately rebels against such things as warped crankcases and misalignment of crankcase bearing saddle bores, bowed crankshafts, out-of-round connecting rod bores, incompatible oils, and the like.

Unfortunately, the manifestations of these things are expressed in the appearance of the bearings themselves, and it seems to be usual to question the metallurgy of the bearing rather than to delve a little into the mechanical conditions surrounding the bearing.

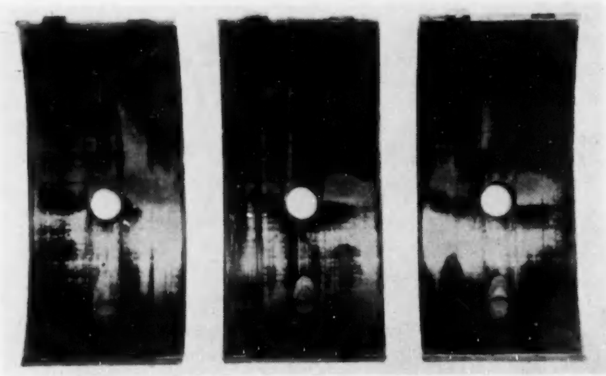
I feel sure that some of the criticism of copper-lead bearings has been without foundation. For example, in Fig. 17 is shown a copper-lead bearing which is typical of a set sent in for examination to determine the cause of failure. No cracks exist; no wear is measurable; and there is no corrosion, no wear nor other indications of distress. The bearing surface has a slight, but healthy and



■ Fig. 24 – Engine test to determine rate of copper-lead bearing weight loss due to corrosion and wear



■ Fig. 25 – Engine test to determine rate of copper-lead bearing weight loss due to corrosion and wear



■ **Fig. 26 – Bearing surfaces discolored by deposits of oxidized oil, sludge, and so on, washed into oil pan from the crankcase interior by lubricating oil with high detergent properties**

natural, coating of an orange-colored lacquer and a few spots where this coating has merged into a dark brown (also natural) but evidently, since this group of bearings did not look exactly like tin-base babbitt bearings would have appeared after similar mileage, they were judged failures and removed from the engine.

The copper-lead bearing shown in Fig. 18 is a similar case. Again there are no cracks, no corrosion and no wear; but, unfortunately, the loaded area carries a surface deposit of an oily black shiny lacquer. The deposit is not harmful, but nevertheless the bearing and its mates were scrapped.

The metallurgy of copper-lead bearings has been widely discussed and is out of place here, but it is interesting to note that there are three general classifications of this type of bearing:

The photomicrographs in Fig. 19 represent a structure range found in bearings used in aircraft engines. The two micros were prepared and photographed in different laboratories which accounts for the difference in sharpness.

The micros in Fig. 20 show a range of structures found in automotive engine bearings. The structures are much coarser even though the chemical analysis of all four specimens shown corresponds to SAE Specification No. 48.

Fig. 21 shows the structure and associated natural surface condition in certain automotive engine bearings. The chemical analysis of the copper-lead lining also follows SAE Specification No. 48.

I am not going to attempt to define which is best; all three classifications have their useful points.

(Note: These photomicrographs were selected from bearings made by all recognized bearing manufacturers and are not from the production of any individual concern.)

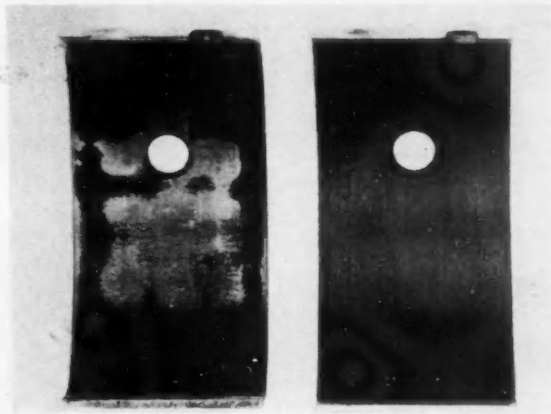
There seems to be a natural antagonism between the raw metal surface of a new copper-lead bearing and the raw surface of a new crankshaft. It is also natural for a copper-lead bearing surface to become coated with a deposit of varnish or lacquer ranging in color from orange to brown and even into a bright shiny hard oily black. This deposit seems to act as a mediator and its action as well as certain characteristics of copper-lead bearings are rather well illustrated in Figs. 22, 23, and 24.

The test represented by Fig. 22 was set up in a 6-cyl engine using copper-lead bearings having a structure similar to specimen *X* in Fig. 20.

A specially selected SAE 30 oil was used and the oil sump temperature was maintained at 275 F. By the use of fusible plugs, the bearing surface temperature showed approximately a 50 F rise. Engine speed was 3400 rpm; duration of test, 100 hr. The engine previously had been used for a number of other tests, and no attempt was made to clean up the interior surfaces to the extent of reaching chemically clean metal.

At each 5-hr interval, the bearings were removed, wiped clean and weighed. The "loss-in-weight" figures represent the average loss per shell. These bearings suffered severely in the first 10 hr after which the weight loss became stabilized up to the conclusion of the test.

In the test represented by Fig. 23, bearings having a structure similar to specimen *Y* in Fig. 19 were used. Exactly the same test routine was followed in the same engine, except that the test was stopped at 75 hr. The same characteristic of high weight loss is found although it occurred a little later on in the test (at 30 hr instead



■ **Fig. 27 – Copper-lead bearings with harmful surface deposits and associated corrosion**

of 5). The same leveling off after the period of distress is shown.

In the test represented by Fig. 24, bearings having a structure similar to specimen shown in Fig. 21 were used. Again we have the same characteristic of high weight loss which, in this case, occurred at 10 hr and then leveled off to a uniform rate until the conclusion of the test.

This temporarily high wear rate is a serious matter, but means of controlling it are available as indicated by Fig. 25.

The test routine exactly followed that of the preceding tests – the same engine was used and the weight loss rate is uniform. The means of accomplishing this reduction in wear are not immediately and universally available but no unusual difficulty is involved.

In Figs. 17 and 18 were shown examples of copper-lead bearings which had been scrapped without just cause, but the scrapping was due to unfamiliar (in so far as the individual inspector was concerned) surface coloration. I have stated that it is natural for a copper-lead bearing surface to become coated with a deposit of varnish or lacquer which affects the surface color. It is possible generally to catalog three types of copper-lead deposits which affect surface color:

1. A *hard,* bright, shiny, oily, black surface – it starts in with an orange color, ages into brown, then often into black. This type of deposit in its orange and brown stages, at least, is natural and has a beneficial effect on the bearing.

Even in its black stage, this type of deposit is difficult to connect with real trouble. Usually the bearing surface is not uniform in color. Certain areas may be dark brown and others black, while some areas show the normal bronze color although overlaid with a light-colored deposit. The discoloring deposits can be removed satisfactorily by immersion for 15 to 30 min in a boiling solution of Oakite No. 29 and water, using 10% of Oakite by volume. After removing bearings from the solution, wash the bearing surface with mechanics soap (such as "Whiz") and water, rubbing briskly to remove the remaining loosened deposits. Rinse, dry, coat with oil and reassemble – providing of course there are no fatigued areas where the lining is completely broken away from the back.

A five-cent ink eraser also does a pretty fair job of removing the deposits.

2. A *soft,* dull, black deposit of the character of dried-out sludge. The deposits usually are not uniform over the bearing surface. Certain areas may be heavily coated or perhaps stained black, while other areas retain their normal color. This condition often can be associated with the use of highly detergent oils in an old engine.

The bearings in Fig. 26 illustrate this point. I have no evidence to show that this type of deposit is harmful, and it may be present on any other type of bearing – that is babbitt or cadmium-silver-copper. This type of deposit can be removed satisfactorily by the means previously described in (1).

3. A *hard, dull,* black deposit (lead and copper sulfide) which is definitely associated with corrosion, and which may cover the entire bearing surface or only parts of the surface. Examples of this condition are shown in Fig. 27. The blackened surface is covered by small pits or cavities which were originally occupied by lead.

The lead has been attacked by certain petroleum acids formed in the oil, converted into lead soap and washed away, leaving a porous copper surface. The formation of the harmful acids is usually the result of excessively high temperature operation (above 260 F oil-sump temperature).

A similar condition of surface pitting (corrosion and selective lead loss) – with and without the black deposit – may be obtained when good oils are contaminated by certain commercially obtainable additives or oiliness concentrates. The danger of using some of these things has been discussed for the last 10 years but the technique of where and when to use some of them is still far from perfect.

A set of beautifully corroded copper-leads recently was sent in for examination. They had been used with a good oil from a standpoint of oxidation resistance, but the operator made it a practice to add 25% by volume of a concentrate guaranteed to prevent valve- and ring-sticking. Corrosion tests of the blended lubricant with three types of copper-lead bearings, cadmium-silver-copper bearings, and tin-base babbitt showed the results recorded in Fig. 28. The criterion of damage is bearing loss in weight.

The corrosion-test routine and equipment will not be described but the results are correlated to practice to the extent that, if the loss in weight exceeds 0.001 g per sq in. in the test, field trouble can be expected. All three types of copper-lead bearings showed very severe weight losses which clearly indicated the expectation of field trouble. It is interesting to note that cadmium-silver-copper and tin-base babbitt bearings were not harmfully affected.

I do not want to imply that all oiliness concentrates, or whatever they may be called, are harmful, but certainly due care must be exercised in their selection for heavy-duty engines where copper-lead or cadmium-alloy bearings are likely to be found.

There is another condition which is sometimes associated with the hard, dull, black deposits and surface pitting, and that is the attack of other types of acids which may be formed under conditions of low-temperature operation, heavy blowby, poor crankcase ventilation, excessive idling, and perhaps a fuel peculiarity which affects the copper rather than the lead. Without a careful laboratory examination, it is very difficult to determine the difference between high-temperature corrosion with selective lead loss and low-temperature corrosion with selective copper loss. However, if the engine operating conditions are known, together with the lubricant and its change schedule, an accurate analysis is possible.

Bearings having the No. 3 type hard, dull, black deposit and its associated lead or copper attack as indicated by small pits and cavities are usually beyond redemption.

Some means are obviously required for easily distinguishing between bearings having the harmless deposits described as Nos. 1 and 2, and the harmful No. 3. The Oakite immersion described seems to offer a fairly practical guide as the No. 1 and No. 2 deposits can be removed nicely while the No. 3 is not noticeably affected. (Potassium-cyanide pickling solution will clean up the No. 3, but the bearing is lost anyway and potassium cyanide is hardly safe to have around a machine shop.) Bearings afflicted by No. 3 have usually been so weakened by corrosion that parts of the lining have disintegrated and broken away from the back structure.

Figures quoted represent loss in weight per sq. in of bearing surface area

Temp	Copper Lead Specimen Type A	Copper Lead Specimen Type B	Copper Lead Specimen Type C	Cadmium Silver Copper	Tin Base Babbitt
225°F	.0034	.0032	.0034	.0009	.00004
250°F	.0066	.0023	.0052	.0022	.00050
300°F	.0174	.0157	.0091	.0007	.00040

■ Fig. 28 – Standard 48-hr corrosion test

■ Rules for Installing Copper-Lead Bearings

1. The crankshaft must be of adequate hardness. For use with bearings of the composition of SAE 48 copper-lead, a crankshaft hardness of at least 300 Brinell is desirable.

2. There must be ample radial oil clearance – not less than 0.001 in. per in. of shaft diameter – and, to be sure of this clearance, the crankshaft diameters must be known accurately. Ample end clearance at the thrust bearing must also obtain.

3. If interchangeable (precision) type main bearings are to be used, the crankcase bearing saddle bores must be round within 0.002 in. and in true alignment to the extent that an aligning bar ground 0.00075 in. under the case bore diameter can be turned by hand with the aid of a 15-in. pipe extension after the caps are tightened down over the bar. If the saddle bores are out-of-round and not in alignment which will permit this, bearings must be align-bored in place so that the discrepancies can be compensated for. The align-bored finish must be smooth, as obtained with a 0.002-in. feed per revolution using a tool bit having a 90-deg nose with the sharp point stoned off. Desired profilometer reading of finished surfce – 30 to 40 micro-in.

4. If interchangeable (precision) type connecting-rod bearings are to be used, the rod bore must not be more than 0.002 in. out-of-round. If the out-of-roundness exceeds 0.002 in., the bearings must be finish-bored to the correct size in the rods. The bored finish must be smooth as described in Item (3).

5. Avoid cap misalignment sidewise by using wrench sockets of the proper diameter.

6. Tighten all bearing bolts and nuts with a torque wrench to uniform settings (as given by the wrench manufacturer or in the Federal-Mogul Shop Manual).

7. Oil gage pressure must be up to original specifications.

8. Engine water jackets, radiator and hose connections must be free and open to insure normal cooling water temperatures.

9. If a reground crankshaft is used, the journal and crankpin surfaces must be ***nice and smooth*** which means ground and lapped. I want to stress this point because the public press has lately been carrying some stories of how good were "rough" crankshafts. These stories are likely to be misinterpreted and a really rough surface finish applied, which would be disastrous.

10. All engine oilways must be thoroughly cleaned out.

11. Connecting rods must be in correct alignment. The crankpin bearing bores and piston-pin bushing bores must be parallel within 0.001 in. in 6 in. and the twist between these bores must not exceed 0.0001 in. to 6 in. The piston gliding surfaces must be square with the axis of the connecting-rod bore.

12. Coat the surface of a new bearing liberally with a heavy engine oil (SAE 50); also the crankshaft surface at assembly. In a newly rebuilt engine, inspect the bearing installation with a pressure tank which will force oil at a predetermined pressure through the lubricating system. The end leakage at the bearings can be observed; all bearing surfaces will be well lubricated before the engine moves under its own power; and a great deal of dirt which might damage the bearings will be flushed out.

13. Select a lubricating oil on the basis of competent engineering advice and then maintain it properly. There is no lack of suitable lubricants.

14. Break in a rebuilt engine with the same routine and care as a new one.

15. When the time for periodic inspection arrives, do not become worried if the bearing surface shows some discolorations and deposits. Determine their nature as explained, clean up the sound bearings, and reassemble them.

Rolling Resistance of Pneumatic Tires

(*Concluded from page 39*)

of the general aspects of the power consumption of tires in their relation to car economy, the writers greatly appreciate the opportunity afforded by the Passenger Car Activity Committee to collaborate in bringing together, organizing, and presenting them in a way which it is hoped will be of interest to the Society and to the industry.

DISCUSSION

Emphasizes Effect of Tire Design

—W. E. Zierer
Chrysler Corp.

THIS paper presents some interesting data relating to the power consumption of tires and the effect of certain operating conditions and design trends.

In general, the conclusions coincide with our test results except that, as stated in the paper, values obtained on laboratory rolls are higher than those checked on the road. All of the data presented were taken on rolls, whereas our tests have been run entirely on the road. A supplement or discussion covering road tests would, in my opinion, make a worth-while addition.

One item which may be questionable is the matter of rim diameter. The authors state that "the power consumption of 15-in. rim tires is from 90 to 95% of its value for 16-in. rim tires." Although we do not have any comparable data on different rim sizes, it would seem that the larger diameter would have less deflection and consequently less drag.

I also have wondered why the effect of tire construction is not discussed except for the mention of six- versus four-ply tires and rayon versus cotton cords.

Our experience indicates that differences in tire design and construction are very important from the standpoint of fuel economy. We have tests showing from 4 to 9% difference in power loss and 3½% difference in economy on tires of the same size, number of plies, and cord material. Since rolling resistance is closely related to tire deflection, a discussion of tread design, tread hardness, cord angle, and other items affecting deflection would seem to be in order.

Airplane Quality Control

CONTROL of quality during the manufacture and assembly of our modern, high-performance military airplanes is without question of paramount importance. "Quality" in itself is a rather intangible property that creates for a product, whether it be airplanes, radios, or roller skates, its reputation. To have a good reputation as a high quality product, a device must comply with the specification requirements of the customer, give entirely satisfactory service, be dependable and, in short, "deliver the goods." Specifically, the most important characteristics of a military airplane from a quality control standpoint are as follows in the order of their importance:

1. Safety
2. Performance
3. Interchangeability

There are other items, but they can be considered of secondary importance under present emergency conditions.

Excerpt from the paper of the same title by J. W. Dunn, director of quality, Curtiss-Wright Corp. (Airplane Division), presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Nov. 1, 1941.

Selection of EARTH-MOVING TIRES for Better Performance and Economy

by WALTER LEE

Tire Design Division, The Goodyear Tire & Rubber Co.

DURING the past five years, great strides have been taken in the development of large tires to carry tremendous loads over rough terrain at speeds up to 25 mph.

Truly these gigantic tires are earth movers. The larger ones weigh over a ton and have carrying capacities well over 15 tons.

In addition to just carrying these unbelievably large loads, these tires must do it without sinking in. They must actually float over the soft, freshly filled-in earth. In other words, a good earth-mover tire must possess the quality of *"flotation,"* as well as the ability to carry these gigantic loads.

Maximum resistance to cutting and abrasion, good traction, and the ability to withstand heat are all qualities to be incorporated for certain types of earth-moving jobs.

It is not advisable or economical to have all these qualities built into one tire to the maximum degree.

Therefore, in selecting an earth-mover tire for a particular type of job, consideration must be given to each of the foregoing qualities.

The type of tire finally chosen must incorporate those qualities most needed for the particular kind of earth-moving job to be done.

■ ■ ■

THE use of pneumatic tires on earth-moving equipment is not new. The first ones used were standard truck and bus tires designed for highway service. These types did very well at the start. However, it wasn't long before someone figured out that, if sideboards were put on the trucks, more dirt could be hauled, so sideboards became standard equipment and more dirt was hauled. Everything went along all right until they got into very soft going and then they bogged down. The sideboard man did some more figuring. This time he decided the tires were too hard – they were cutting in too much – so he took some of the air out and they performed much better. They went along fine for quite a while until the tires began crushing out through the sidewall. Then a man came out from Akron and was introduced to earth-moving tires in off-the-road service.

No doubt his initial statement was: "The tires were not designed for that kind of use." However, later on when he talked with the sideboard man and they both did a little figuring, they came out with a new tire quality which they called *"flotation."*

[This paper was presented at the National Tractor Meeting of the Society, Milwaukee, Wis., Sept. 25, 1941.]

From that time on, tires began to be designed especially for the earth-moving type of service. Whatever other qualifications a tire had, if it did not have flotation, it was not an earth-mover.

So, the next milestone was a tire that had flotation plus, but very little else for earth-mover service. The change was made in one jump from the standard-type highway tire to the doughnut-type airplane tire.

This tire provided all the flotation needed, but failed from lack of load-carrying capacity in heavy service. The inside bead diameter was too small for the cross-section and, therefore, the fabric cords in the carcass did not have enough anchorage to hang on. The doughnut-type tire had good flotation, but was a poor load carrier.

The next logical step was to keep the large cross-section for flotation, but to increase the rim diameter to get a better-proportioned tire for heavy loads.

This brings us up to the present earth-mover tire proportions of which the 18.00-24 is a good example.

Fig. 1 shows the complete engineering data for the 18.00-24 size. Notice that the ratio of outside tire diameter – 62.30 in. – divided by the rim diameter – 24 in. – is 2.6. This is well below the ratio of 3.2 which marks the begin-

ning of the "doughnut" type. The doughnut tire has many uses, but not in the heavy-duty earth-moving field.

One other ratio that is well proportioned in the 18.00-24 size is the relation between the number of plies in the carcass, or body of the tire, and the recommended inflation pressure. This relation is roughly expressed for all ordinary purposes as:

$$\frac{\text{Section width} \times \text{inflation pressure}}{\text{Twice the number of plies}} = \text{Fabric stress index.}$$

For the 18.00-24 16-ply tire this figures –

$$\frac{20.00 \times 40}{2 \times 16} = 25$$

When this ratio or index goes much above 30 for earth-mover tires, you can expect carcass trouble, especially when the tires get into rough going, such as over terrain with a great deal of rock mixed in with the loose dirt.

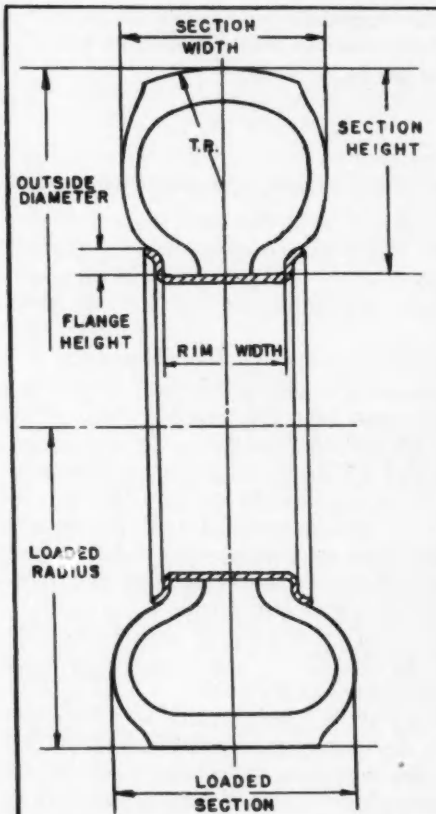
From the foregoing relation between the number of plies and the air pressure, it is easy to see what happens when an operator tries to compensate for an overload by increasing the air pressure. He figures that, since the air carries the load, simply increase the pressure when you are hauling more dirt.

He fails to take into account that, while the air carries the load, the tire carries the air.

A certain amount of adjusting inflation pressures to meet various loads under special operating conditions can be done. However, it is best to get the advice of a tire man first. If larger loads must be carried, go to extra-ply tires

■ Fig. 2 – Earth-mover tire suitable for trailing wheels on tractor-drawn scrapers

ENGINEERING DATA



SIZE 18.00-24

TYPE EARTH MOVER

NO OF PLIES	12	16	20
MAX. INFLATION	30	40	50
MAX. LOAD-25 M.P.H.	10230	12130	13800
MAX. LOAD-10 M.P.H.	11460	13580	15450
RIM WIDTH	13.00	13.00	13.00
FLANGE HEIGHT	3.00	3.00	3.00
DUAL SPACING	22.75	22.75	22.75
OUTSIDE DIAMETER	62.30	62.30	62.30
SECTION WIDTH	20.03	20.03	20.03
SECTION HEIGHT	19.15	19.15	19.15
LOADED RADIUS	27.92	27.92	27.92
LOADED SECTION	21.55	21.55	21.55
CONTACT AREA	292.00	292.00	292.00
TREAD RADIUS	19.50	19.50	19.50
TREAD WIDTH	19.20	19.20	19.20
TIRE WEIGHT	520.00	577.90	633.90
TUBE WEIGHT	58.30	58.30	58.30
FLAP WEIGHT	11.30	11.30	11.30

THE GOODYEAR TIRE & RUBBER CO.
AKRON OHIO DATE 9-11-41

■ Fig. 1 – Engineering data on 18.00-24 earth-mover tire

■ Fig. 3 – Typical short-haul dirt job using tractor-drawn scrapers with earth-mover tires

before increasing the air pressure. In those cases where the equipment will permit oversizing, the next larger tire will be the answer to the heavier loads.

This paper so far is a record of the early efforts in searching out the best size and general characteristics of a tire most suited for carrying heavy loads over rough terrain where, most of the time, there is no semblance of a road. What changes will be made in the next few years are hard to forecast. However, there are still a great many problems to be solved before the earth-mover type of tire reaches the plane of satisfaction now enjoyed by the standard highway truck tire.

Now then, in selecting a tire for earth-moving jobs, it is necessary to know three things:

1. What load is to be carried?
2. What type of terrain – loose dirt, rock, haulage roads?
3. What type of equipment – tractor-drawn or self-powered?

The first question is obvious. The second and third really should be grouped together into: "What is the nature of the job to be done?"

In general, earth-moving jobs can be divided into three main types, each of which calls for a separate tread design for the best operation:

■ Fig. 4 – Another short-haul dirt job using earth-mover tires on free-rolling wheels

■ Fig. 5 – Tractor-drawn scraper spreading its load at the fill

■ A – Short Hauls

The first and most common type of earth moving is excavating and moving loose dirt over rough terrain where the length of haul between the cut or borrow pit and the fill is short – a few hundred yards. The equipment will be a tractor-drawn scraper with free-rolling wheels and a speed of about 4 to 6 mph. Therefore, the tires will not be subjected to driving torque or braking action, nor will they be subjected to smashing impact blows since the speed will be low.

■ Fig. 6 – Mud-grip or bar-type tire suitable for driving wheels on self-powered earth-moving equipment

The prime requisite for these tires will be plenty of flotation, that is, a large contact area. Since loose dirt won't support unit pressures much over 50 psi without deep penetration, earth-mover tires must be large enough in cross-section and outside diameter to develop a contact area of about 20 sq in. per 1000 lb of load to be carried. They must also have a tread pattern suitable to resist cutting. Such a pattern would be an overall design with shallow buttons closely nested to give full protection to the carcass or the body of the tire. Fig. 2 shows this type of tire; Figs. 3, 4, and 5 show this type of job.

■ B – Long Hauls

Where the ground conditions are the same as just described, but the earth must be hauled long distances, say a half-mile or more, the equipment, in most cases, will be self-powered scrapers, or tractor-trailer units.

The speeds on these jobs are relatively fast, up to 25 mph. The tires must take both the driving torque and braking action. For a tire to stand up on these jobs, it must have just about everything.

In addition to having plenty of flotation, being a good high-speed load carrier, standing up under stresses and strains of driving and braking, it also should have a tread pattern with maximum traction to keep the unit going through deep and slippery mud.

Unfortunately, this quality of traction is usually obtained at the expense of tread wear and cut resistance. To get mechanical traction we must spread apart the tread pattern and, when we do this, we expose the carcass of the tire to cuts and bruises. These open-tread patterns are called "bar-type" tires. They permit the tire to grip and to pack down large slabs of dirt or mud. This mechanical gearing action permits the tire to pull through mud holes in which closely nested tread patterns would be mired.

These tires usually have one-way tread patterns and must be mounted on driving wheels to rotate in the direction to plow the mud out through the open-end channels to each side of the track.

Rotating in the opposite direction on driving wheels, the slabs of mud would be plowed in toward the center of the contact area and thereby pocket or pack in and clog the tread pattern – then, on the next rotation, the tread surface would be solid with mud and would have no more traction than a smooth tire.

When these "one-way" tires are used on free-rolling wheels, or on trailing wheels with brakes, they are mounted to rotate against the directional arrow to get the best tread wear.

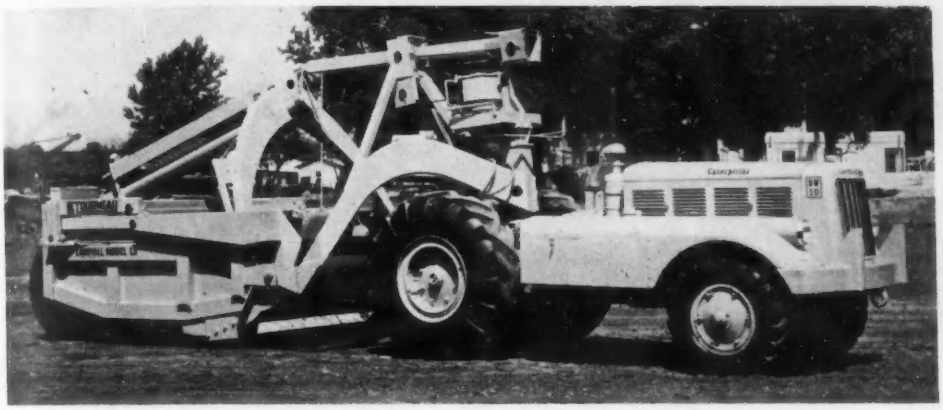
At this time I would like to point out that the industry is not of one mind with respect to these open-bar-

■ Fig. 7 – Start of a long haul on an earth-moving job showing tractor-trailer trucks with bar-type mud-grip tires

type directional tires. They have their greatest use on the lighter type of equipment where some sort of mechanical advantage is needed to supplement the natural friction between the tire and the ground to obtain the necessary traction.

On extremely heavy equipment where the drive wheels are well loaded, the coefficient of friction between rubber and packed dirt provides ample forward traction for all ordinary going.

■ Fig. 8 – Long-haul job showing "Tournapol" on bar-type tires hauling a 12 cu yd scraper



■ Fig. 9 – Close-up of high-speed tractor-scraper unit equipped with bar-type tires on the driving wheels

On these heavy units, tires with closely nested tread patterns are satisfactory and give a somewhat smoother ride. Bar-type tires, in some cases, are a distinct disadvantage, especially in starting up under full load where there is a tendency to dig in and to bog down in sandy loam if the wheels slip or spin.

It may work out that the bar-type tire will be limited to those cases where there is not enough load on the drive wheels to provide sufficient drawbar for the trailing load – or it may work out that straight friction between the rubber and the dirt will provide ample traction for 90% of the conditions.

On the remaining 10% where mud is the limiting factor, many times the operation will be shut down anyway for other reasons.

If this proves to be the answer, we will be able to eliminate one type of tread design which will simplify the standardization picture tremendously.

However, for the time being, both tread patterns are in general use on driving wheels and no definite conclusions have been reached.

Fig. 6 shows a bar-type tire which represents a tread pattern having satisfactory mechanical traction in slippery going.

Figs. 7, 8, and 9 show the bar-type tire on various types of equipment used for long-haul earth moving.

■ C – Mining and Quarrying

The first two classes of earth-moving jobs are strictly "off-the-road." The third general classification is strip mining and quarrying. These jobs have haulage roads and the equipment may be either straight trucks or tractor-trailer units.

The prime quality that a tire must have for satisfactory service in mines and quarries is maximum resistance to cuts and bruises. In other words, the carcass of the tire must be protected from road hazards in every way possible. Fortunately, traction is not a prime requisite for tires on the hard, rough-surfaced haulage roads. Therefore, we do not need an open tread pattern, but we do need a deep pattern and one that will not only have maximum cut-and-bruise resistance, but also the quality to resist road abrasion.

Flotation, which was of prime importance on the two previous classifications of earth-mover jobs, is now of secondary importance for a good mine-and-quarry tire. There is no tire penetration on haulage roads. However, the tire with the greater flotation will be the easier on the roads. The less unit pressure between the tire and the road, the less road maintenance will be necessary.

So, while tire flotation is of secondary consideration on these jobs, it is not to be disregarded entirely.

The other tire qualities, such as being a good high-speed load carrier, ability to take driving torque and braking action, and so on, must be taken for granted to start with. No tire can qualify in any phase of the earth-moving business if it has not the ability to carry tremendously heavy loads.

The reference to moving the earth assumes that fact. Iron ore, coal, and stone are just heavy dirt as far as the earth-mover tire is concerned.

■ Fig. 10 – "Rock" tire suitable for use on trucks in strip mines and quarries

In their ability to resist and retard cuts from rocks and other sharp objects, tires for mines and quarries have come to be called rock tires – with trade names such as "rock service," "rock grip," "hard rock tires," and so on. Fig. 10 shows a rock-type tire suitable for mine-and-quarry work. Fig. 11 shows a close-up of a typical strip mine operation. Fig. 12 illustrates a hard-rock quarry job.

Naturally, haulage roads are very abrasive. Ordinary tread rubber that would be quite satisfactory on smooth highways would wear down quickly on haulage roads made of iron ore and rock. Also, a good mine tire must have a tread pattern of such a nature that a maximum of rubber bars and a minimum of tread grooves are presented to the ground for road contact.

By having the sides of the narrow tread grooves slanting and well rounded, the carcass is protected against road hazards to the maximum degree and the tread still retains sufficient traction, even though this quality is secondary.

Mine-and-quarry tires get very hot on certain types of operations where the road grades are steep for long distances. A great deal of heat is generated by the brake drum on the down trip. Much of this heat is conducted through the rim to the beads of the tire. We have had cases where the lower sections of the tire were completely devulcanized due to brake-drum heat in less than 400 hr of service. The only way that this problem was licked on these particular operations was by installing a spray system to water-cool the brake drums. This extreme is not encountered very often, but it goes to show what can happen. Heat is the worst internal enemy that a tire has.

We have now described the three major classes of earth-moving jobs and the selection of the type of tire best suited for each need.

After knowing the kind of tire to be used, next comes the cross-section, outside diameter, rim diameter, and number of plies, all of which are directly connected with the load to be carried. The load information is usually furnished by the equipment engineer.

To crystallize some of these general statements about the selection of tires for earth-moving jobs, let us take a definite case where we have been asked for the best tire size on a new piece of equipment. This equipment is still in the drawing-board stage and is to be a tractor-drawn scraper with a struck capacity of 20 cu yd (heaped 24 cu yd). Sixty-five per cent of the gross load is to be carried across the rear axle. The estimated weight of the equipment is 30,000 lb.

Since it is a tractor-drawn unit, the wheels on the scraper will be free-rolling. Therefore, the type of tire will be the standard earth-mover design with a shallow-button overall tread pattern, such as shown in Fig. 2.

The heaped capacity of 24 cu yd at an average unit weight of 2200 lb per cu yd figures a payload of 52,800 lb. This value, added to the empty equipment weight, gives a gross load of 82,800 lb.

With a 35-65 load distribution, dual tire assembly must be used on the rear if we keep the same size tire all around. However, since this is not absolutely necessary, we were asked to figure the unit both ways. First, with dual rears and single fronts – second, with single tires all around.

The tire man would much rather recommend the larger single tire than the smaller dual tire for off-the-road service. With dual assembly, about 10% of the time one tire is forced to carry the whole load. This condition is especially true over rough terrain such as frozen ground or cut-over areas, where short tree stumps are numerous.

With dual rears and single fronts, all of the same size, the load will be slightly greater on the front tires with

■ Fig. 11 – Dump truck on an iron-ore strip-mining operation using rock-type tires on the rear

35-65 ratio; 35% of 82,800 lb is 28,900 lb. The axle load divided by 2 gives a front-tire load of 14,450 lb.

Referring to the Tire and Rim load tables we have three tires that will handle the load:

16.00-32 20-ply at 55 psi inflation.

18.00-24 20-ply at 45 psi inflation.

21.00-24 16-ply at 30 psi inflation.

Since loose dirt will not support unit pressures much above 50 psi without exceedingly deep penetration, the 16.00-32 tire is out because of lack of flotation. It develops less than 225 sq in. of ground contact area. Because of its large rim diameter it is a good load carrier but, for 14,450 lb, it is necessary to inflate the tire to 55 psi to keep away from excessive sidewall flexing.

For all practical purposes we can assume that the average ground pressure under contact is about 12% higher than the tire inflation pressure. This means an average ground pressure of over 60 psi for the 16.00-32 tire and loose dirt won't support that load satisfactorily. We need a tire with more flotation.

At 45 psi inflation and 14,450 lb load, the 18.00-24 20-ply earth-mover tire develops a gross contact area of about 290 sq in. which, in turn, demands an average ground pressure of almost 50 psi for the load in question. Under most conditions, loose dirt will support this pressure. However, the 18.00-24 tire will have nothing to spare in the way of flotation.

To have a good factor of safety, especially for overloads, the 21.00-24 16-ply tire inflated to 30 psi would be the better recommendation for dual assembly with the same size tire all around.

Now then, if we get away from the dual assembly, the 21.00-24 16-ply will still be the best tire on the front, but a much larger single tire must be used for the rears.

Sixty-five per cent of the total load of 82,800 lb is 53,820 lb – and that divided by 2 equals 26,910 lb for each rear tire. The most reasonable recommendation from the load-inflation tables is the 24.00-32 24-ply earth-mover tire inflated to 45 psi. This tire develops a ground contact area of about 520 sq in.

This size does not have quite the load-carrying capacity desired and also the flotation is slightly on the low side. Nevertheless, since a tire in a single assembly has a greater factor of safety than in dual assembly, 2 single 24.00-32 24-ply tires on the rear and 2 single 21.00-24 16-ply tires on the front will give better all-around tire service than any dual arrangement.

All three possibilities will be presented to the equipment manufacturer:

	Front Tires	Rear Tires
First choice –	21.00-24 16-ply	24.00-32 24-ply
Second choice –	21.00-24 16-ply	21.00-24 16-ply (duals)
Third choice –	18.00-24 20-ply	18.00-24 20-ply (duals)

If cost is of primary importance on this piece of equipment, certainly the 18.00-24 tire will be selected.

If initial cost isn't the all-important consideration, the decision will be between the six 21.00-24 tires and the two 21.00-24 fronts with 24.00-32 rears. The cost of each of these selections will be about the same.

■ Fig. 12 – Rock-type tires on an all-rock quarry job

The equipment engineer would like the six 21.00-24 tires. This will give him one size to deal with in respect to tires, wheels, rims, and so on. Also it will reduce the number of spares to be kept by the customer.

The tire engineer, however, would like the single tire assembly with 21.00-24's on the front and 24.00-32's on the rear. He knows that a great deal of the tire trouble in "off-the-road" service is linked up with the use of dual tires. Too much of the time one tire is forced to carry the whole load.

The final decision, of course, will rest with the equipment manufacturer.

What I have said sums up roughly the process of analysis and elimination that the tire design man goes through when he is asked for a tire recommendation. It is impossible to get just exactly the right size tire for every piece of equipment. Considerable judgment must be used. He must take into account all the peculiarities and possibilities of the particular equipment and job to be done, and then strike a practical and economical average for his recommendation.

Whether or not he was correct will be pointed out clearly in the following months by the performance of the tires in actual operation.

TWO-SPEED Supercharger Drives

by F. M. KINCAID, JR.
Wright Aeronautical Corp.

SHORTLY after the first World War, engine manufacturers turned to the production of supercharged engines to increase the power output. Obviously the problem was to increase the quantity of air taken into the engine, as the power is directly proportional to the mass of the air charge. This increase in charge can be attained by means of a blower with a smaller engine-weight increase than it can be done by increasing the piston displacement. All of the early mechanical superchargers were driven by single-speed gears.

A highly supercharged engine is a delicate thing to handle. This type of engine must be operated at part throttle at low altitudes; at full throttle the supercharger will supply more air than the engine can safely use. The heat of compression also heats the fuel and air mixture excessively, reducing the power and inducing detonation.

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Oct. 30, 1941.]

This quickly causes the spark plugs, valves, pistons and rings to fail. At the higher altitudes, the atmospheric temperature and density have declined so much that the engine can safely be operated at full throttle.

European manufacturers first concentrated on engines to develop their maximum power at high altitudes for interceptor fighters. The primary objective of American manufacturers was to develop engines with high power at "take-off" for heavily loaded transports and bombers. For this reason the ratio of blower speed to engine speed tended to be higher in European engines than in American engines. European operators soon demanded more power for take-off. American operators demanded more power at altitude.

A SINGLE-SPEED supercharger with a compression ratio adequate for maximum power at high altitudes will supply more air than the engine can safely use at low altitudes. Two-speed supercharger drives are provided to obtain high power for take-off at low altitudes and maximum power at high altitudes with one blower.

The Wright two-speed drive utilizes a single compound spur-gear system with the intermediate gears connected by a rotating hydraulically actuated friction clutch for the high ratio, and a planetary reduction gear with a clutch in the low ratio. A later model has the intermediate gears coupled by a roller clutch in low ratio and a planetary step-up gear and stationary clutch in high ratio.

The Pratt & Whitney two-speed drive utilizes hydraulically operated cone clutches for both the low and high ratio. A fluid coupling is used to accelerate the impeller to a speed between that of the high and low ratio drives for the shifting operation only. Two parallel units are used to reduce the gear sizes.

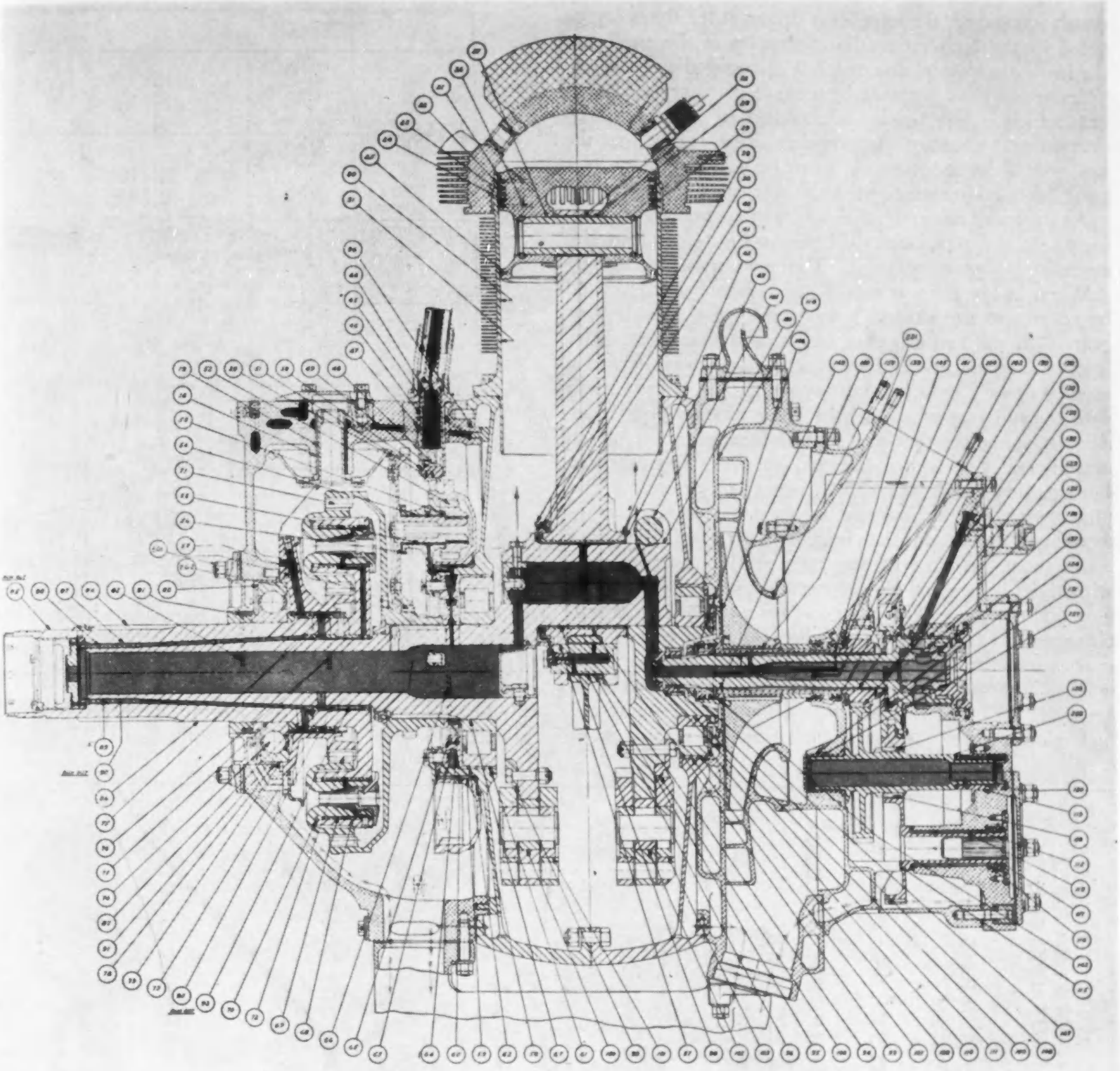
Bristol uses three clutch units equally disposed above the driveshaft so that each unit carries one-third of the load. Each unit contains two multiple-disc clutches, one for each ratio. Oil for the hydraulic actuating cylinders is cleaned by a pair of centrifuges before it enters the clutch units.

The Rolls-Royce supercharger drive uses semi-centrifugal mechanically actuated clutches. One clutch is provided for the low ratio and two similar clutches for the high ratio. The mechanical linkage is operated by a hydraulic cylinder with scavenge oil pressure.

In the Junkers two-speed drive, a pair of bevel gears drive a layshaft connected to the impeller shaft by two intermediate gears. The low ratio intermediate gear is coupled to the shaft by a roller clutch. The high ratio gear is connected by a mechanically operated friction clutch. An aneroid mechanism prevents engagement of the high ratio clutch at low altitudes.

The Mercedes-Benz DB-601 engine has a variable-speed supercharger drive consisting of a straightforward gear system driving the final driveshaft at a speed slightly higher than the highest ratio. The speed is then reduced by a fluid coupling. Operation of the drive is entirely automatic, the ratio of the coupling being regulated by a control unit actuated by an aneroid capsule. The slip of the fluid coupling in low ratio at the lower altitudes causes the oil in the coupling to be heated considerably. This would be a great disadvantage in transport or patrol planes intended for operation at low altitudes. However, this might be an advantage for fighter craft as the high heat rejection would aid in quickly warming up the system while climbing to high altitudes.

The DB-601 design has created considerable interest in this country. We can be sure that the fluid coupling will be studied carefully and possibly adapted to our requirements. Further developments of two-speed drives will be made adapting them to two-stage superchargers with intercoolers between the stages.



■ Fig. 1 – Cross-section of C9GC with single-speed supercharger

American manufacturers then introduced engines with a two-speed blower, adding the high ratio to a moderately supercharged engine. European manufacturers adopted the two-speed drive, adding low ratio blower to their highly supercharged engines.

Both European and American manufacturers now furnish two-speed drives in the same range – from 6:1 to 8:1 for the low ratio for use at "take-off" and altitudes up to approximately 10,000 ft, and from 9:1 to 11:1 for maximum power at the higher altitudes. This type of supercharger will supply the maximum amount of air that can be used by the engine without the adoption of intercoolers. A curve showing the altitude gained by the second speed is given in Fig. 2.

A cross-section of a complete radial engine with a single-speed supercharger is shown in Fig. 1. The supercharger is located just aft of the power section. The single-speed drive is located in the rear section. This general arrangement is almost universal for radial engines. The accessory and supercharger driveshaft, called a tailshaft, is supported at the front by splines in the crankshaft and at the rear by a bearing in the rear cover. The hollow impeller shaft is supported by plain bearings on the tailshaft. Details of a single-speed drive are illustrated in Fig. 3. This drive consists of a tailshaft gear *A*, shown integral with the tailshaft, which drives an intermediate gear *B*, made integral with a larger-diameter intermediate gear *D*. This gear *D*, drives the impeller shaft gear *E*. The ratio for this particular drive is 10:1; though a drive of approximately 7:1 is more commonly used.

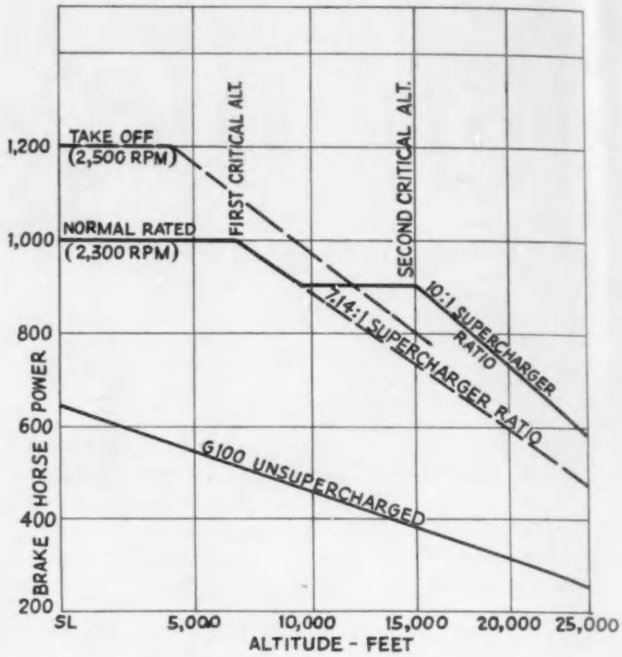
■ Wright Two-Speed Plate Clutch

The Wright two-speed supercharger drive is similar to this single-speed drive except that there is a hydraulically actuated friction clutch between the small intermediate gear *B*, and the larger intermediate gear *D*. When this

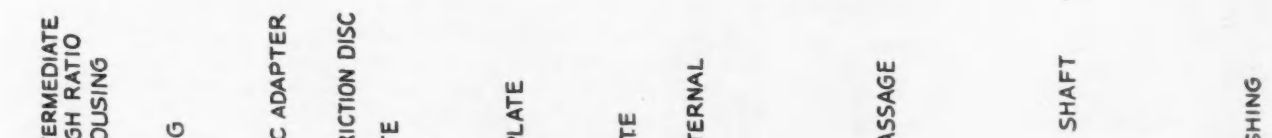
clutch is engaged, the impeller is driven at ten times engine speed in the high ratio. For operation of the engine in the low-ratio blower, this clutch is disengaged and the two gears are coupled together by a small planetary reduction gear which is very similar in construction to the large reduction gear used for the propeller drive, except that the sun gear is designed so that it can be held stationary by a small friction clutch located in the rear cover of the engine.

An exploded assembly of this type of supercharger drive, for 7.14:1 and 10:1 ratios, is shown in Fig. 4. A complete assembly is shown in Fig. 5. The operation is as follows:

When in low ratio as shown, the tailshaft gear (coupled by springs to the tailshaft) drives the small intermediate gear. On the anti-propeller end of this gear there is an integral internal gear. This gear drives five pinions in a small planetary reduction-gear system. The pinions mesh with the sun gear which is held in a stationary position by three clutch discs that are squeezed between the clutch housing and the two steel intermediate plates by a piston in a hydraulic cylinder actuated by engine oil pressure. The pinions in the reduction gear are mounted on a spider integral with the intermediate impeller shaft. As these pinions rotate about the sun gear they carry the intermediate shaft with them at a speed slower than the small intermediate gear (ratio 0.7:1.) This shaft in turn is splined at its propeller end to the large intermediate impeller gear. This gear drives the impeller shaft at a speed lower than it would if the multi-plate clutch within the gear itself were engaged. When it is desired to operate the engine in the high blower ratio, the control valve on the rear cover of the engine is pulled to the rear. This cuts off the oil supply to the low-speed hydraulic cylinder and simultaneously allows the oil in this low-speed cylinder to drain out into



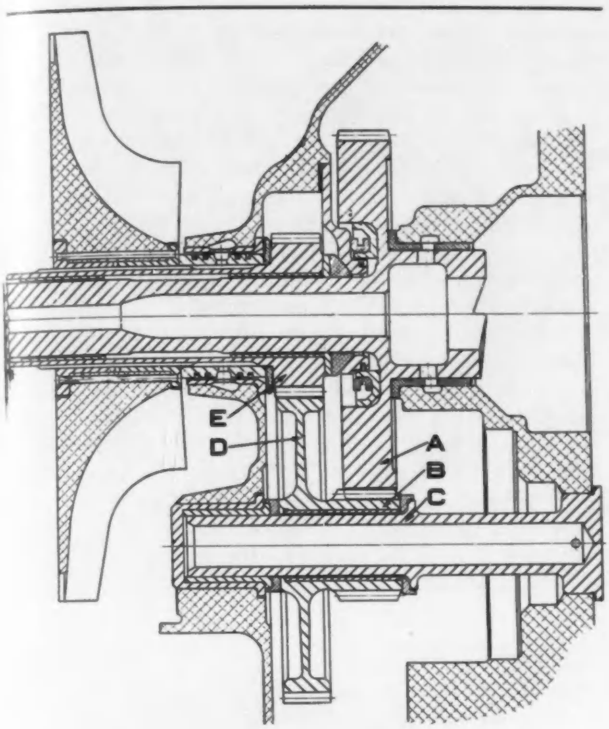
■ Fig. 2 – Altitude gained by a second speed



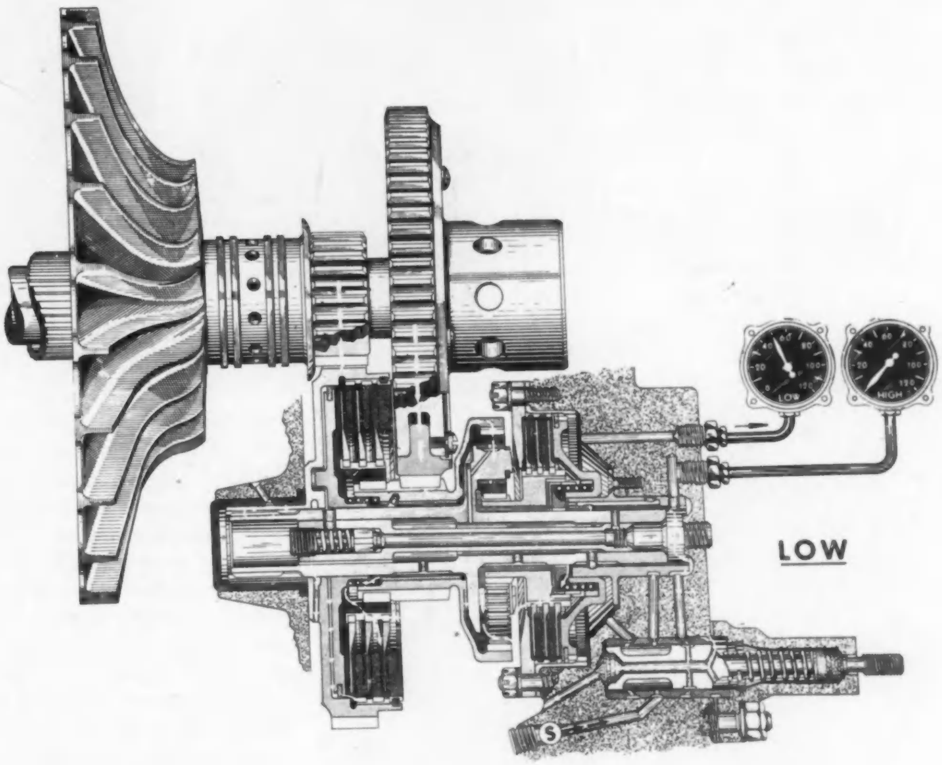
■ Fig. 4 – Wright two-speed clutch – exploded

three clutch discs between the retaining plate fastened in the rear end of the gear and the two intermediate steel plates. This makes the small intermediate gear, meshing with the tailshaft, and the large intermediate gear, meshing with the impeller gear, the equivalent of one solid gear as shown in Fig. 3 for the single-speed drive.

There is one important difference between the high-speed clutch and the low-speed clutch. The high-speed clutch is rotated at 2½ times engine speed when the drive is engaged in the low ratio and at 3½ times engine speed when the drive is engaged in high ratio. The low-speed clutch housing and actuating hydraulic cylinder are stationary at all times. The rotating clutch has one big advantage; the centrifugal force on the oil increases the total pressure on the piston and greatly increases the clutch capacity. It also has one big disadvantage. This was the source of considerable trouble shortly after these clutches were introduced. The high-speed clutch cylinder acts as a centrifugal separator, separating the sludge in the engine oil from the oil. This sludge is not formed in the clutch itself. It is formed in other parts of the engine, in the combustion chamber by charring of the oil and separation of lead from the fuel, or by carbonizing of the oil on the back of the pistons and other portions of the engine that run at an extremely high temperature. These small carbon and lead particles are carried to all parts of the engine. They are extremely small



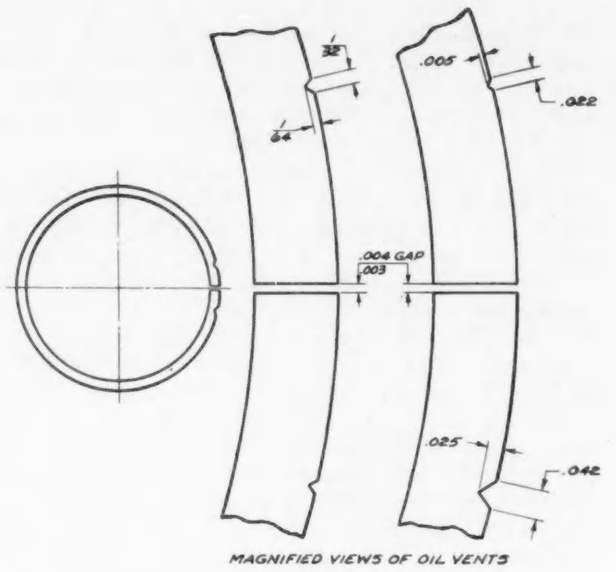
■ Fig. 3 – Single-speed supercharger drive



■ Fig. 5 – Wright two-speed clutch assembly

the rear section of the engine, dropping the pressure to zero. It also uncovers ports in the valve sleeve and allows the oil supply to go through the intermediate shaft and into the hydraulic actuating cylinder housed in the large intermediate gear. When oil pressure is applied in the high-speed cylinder, the piston is forced to the rear and squeezes

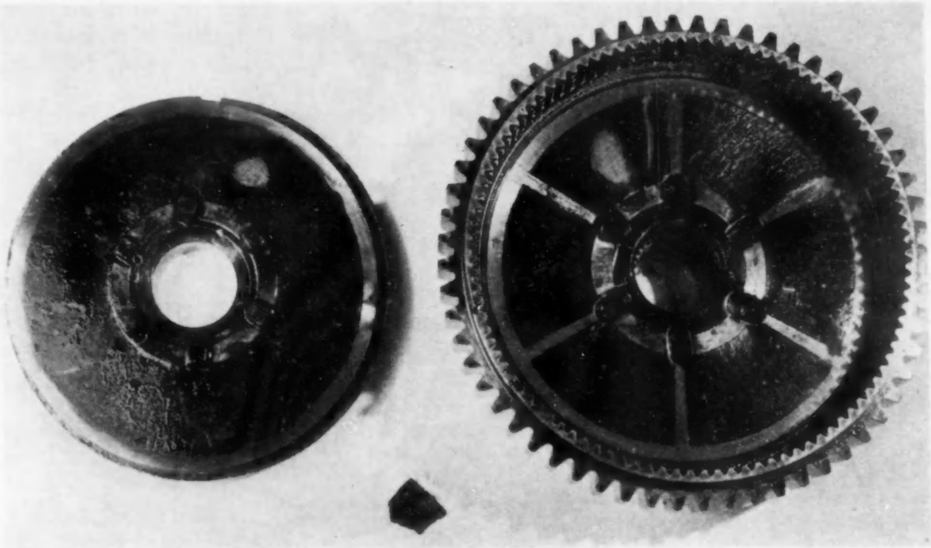
and will pass through the finest filter; but, when they get into the high-speed clutch actuating cylinder and are whirled at approximately 7000 to 8000 rpm, these heavy particles are thrown to the outer parts of the cylinder. The particles become interlocked into a very hard mass very much like hard-packed sand on a beach. After a very



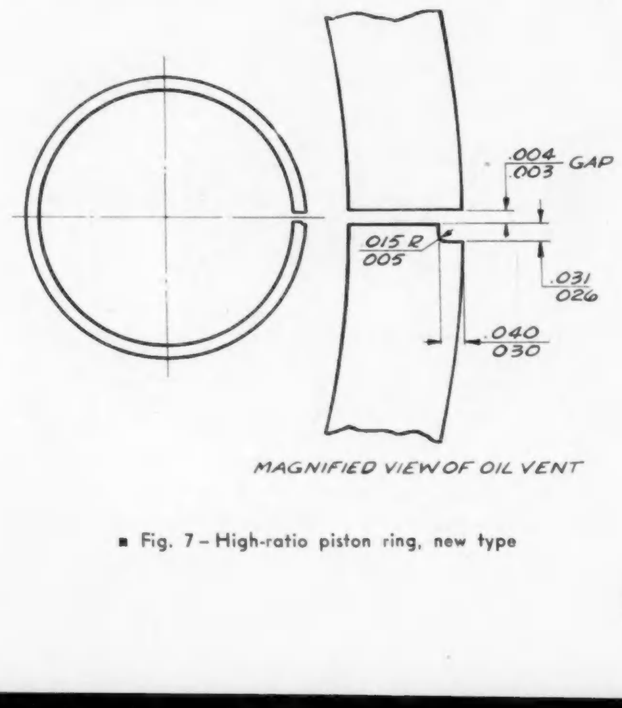
■ Fig. 6 – High-ratio piston ring, old type

long continuous period of running in high blower, this sludge may be packed in so tightly that the spring which returns the piston of the high-speed clutch to the neutral position will not be able to force this sludge out of its path. This clutch then remains engaged when the operator shifts the control valve from high to the low position. But the low-speed clutch is entirely independent of this high-speed clutch and it becomes engaged. Two different clutches are then trying to drive the same impeller at two different speeds and one of the clutches will fail. Invariably one clutch is left to drive the supercharger.

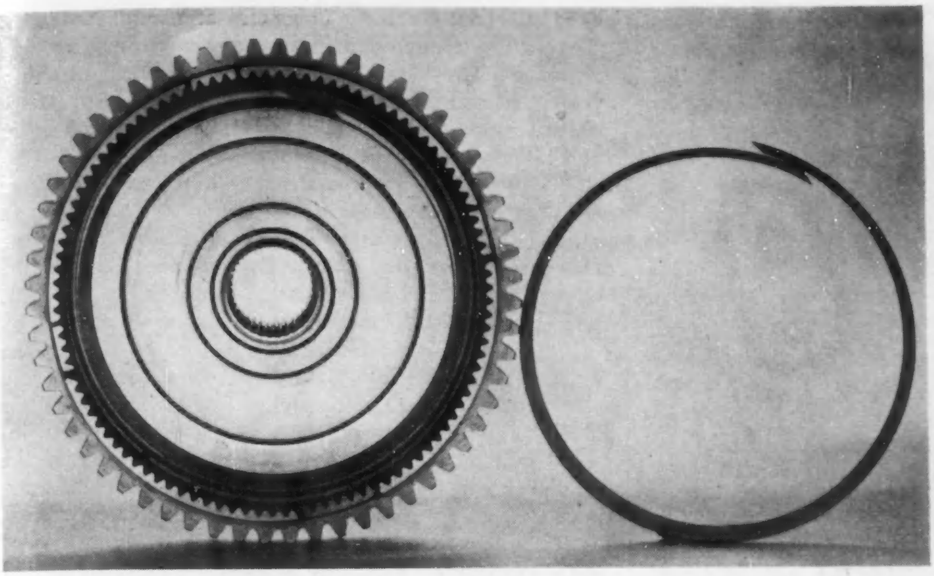
A very minor detail was the main cause of this trouble. This was the type of vent originally incorporated in the high-speed clutch to allow the oil to pass out of the cylinder when the clutch was disengaged. The only way by which this oil can leave the cylinder is through an orifice at the extreme outer diameter of the cylinder. Even if the passage through the central part of the intermediate shaft were wide open, the oil would not flow back out through the same passage by which it enters because of the centrifugal force tending to throw it from the center of the cylinder to the outer diameter. To allow this oil to pass out of the clutch, two small "V" notches placed ½ in. from the gap were provided in the piston ring. These notches were intended to be open all of the time, and at all times while operating in high ratio there is a continuous flow of oil into the high-speed clutch and out through these notches. This type of piston ring is shown in Fig. 6. The original drawing specified that the notches be 1/32 in. wide and 1/64 of an in. deep. As long as the notches were made to this size, the operation was satisfactory; but a fractional dimension like this on any drawing means that the dimension may vary with a tolerance of ±0.010 in. That means that it was permissible to make this ring with two notches 0.022 in. wide and 0.005 in. deep. This was entirely too small a passage to allow the oil to pass out of the clutch after a small amount of sludge has accumulated in the outer diameter of the cylinder. A small amount of sludge can then block off the exit through the notches and the oil remains in the cylinder. With the oil supply to the



■ Fig. 8 (right) – High-ratio clutch cylinder after 477.5 hr of simulated normal operation



■ Fig. 7 – High-ratio piston ring, new type



■ Fig. 10 – Collapsible piston ring

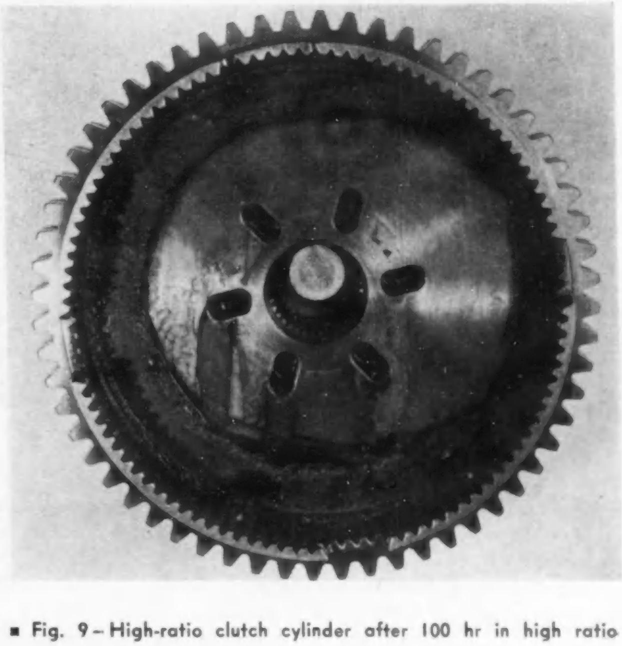
center the oil pressure increases as the radius increases due to centrifugal force on the oil. At the periphery of the cylinder this pressure may reach approximately 175 psi. This means that the average pressure for the whole cylinder is about 80 psi, and this is more than sufficient to keep the high-speed clutch fully engaged. When the control valve is shifted, the low-speed clutch engages about 0.5 sec after the oil pressure is applied to the low-ratio piston. The small ring of sludge and the centrifugal oil pressure in the high-ratio clutch keep it engaged as effectively as a solidly sludged clutch.

To cure this trouble the size of the vent in the piston ring was controlled more accurately. In order to identify this new ring, the two notches spaced ½ in. from the gap were removed and a single notch at the gap was substituted as shown in Fig. 7. This size vent will not be blocked by sludge accumulated in ordinary operation. This new ring promptly eliminated almost all of the sludging trouble encountered with this supercharger drive.

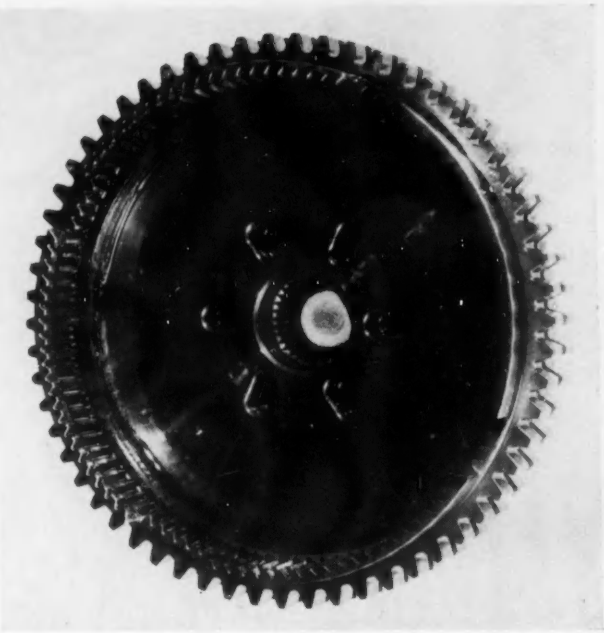
Fig. 8 shows a high-speed clutch gear and clutch housing that was used with the equivalent of this new piston ring and operated for 477:30 hr in an endurance test which simulated operating conditions usually encountered in actual service. This particular ring had very small notches, so small they are invisible in the picture, but it did have a 0.015-in. gap. The new piston ring has a vent twice this size. This operation left the clutch cylinder almost completely free of sludge with only a very small ring of hard-packed sludge in the corner of the cylinder, a quantity too small to do any harm. This test was repeated with equal success with the improved ring.

Sludge will collect consistently in every rotating cavity not vented at close intervals at the periphery. Its angle of repose is from 45 deg to 60 deg. If these characteristics are carefully noted by the engine designer, the sludge will do nothing worse than make a very "messy" and "dirty" engine after long periods of operation. If the designer does not note this behavior, he will surely be disagreeably surprised when his engine is operated for a long period.

Fig. 9 shows a high-speed gear with an extremely bad accumulation of sludge. This clutch ran for 100 hr in high ratio, with only 1 2/3 hr in low in that time, under the worst sludging conditions obtainable in the test laboratory. To avoid sludging under these conditions, a special type of centrifugal valve is used in the high-ratio clutch. This consists of a collapsing piston ring, shown in Fig. 10. At normal engine operating speeds this ring is expanded by centrifugal force and seals as effectively as the older type of piston ring. At engine speeds below the normal operating range, the ring collapses and allows the sludge to be thrown out of the clutch cylinder. Fig. 11 shows a clutch cylinder which was assembled with a collapsible ring, after a 100-hr test under the same conditions that caused the cylinder in Fig. 9 to become jammed with sludge. This cylinder has a small quantity of sludge in the corner. The clutches were in perfect operating condition at the end of this test.



■ Fig. 9 – High-ratio clutch cylinder after 100 hr in high ratio



■ Fig. 11 – High ratio clutch cylinder after 100 hr in high with collapsible ring

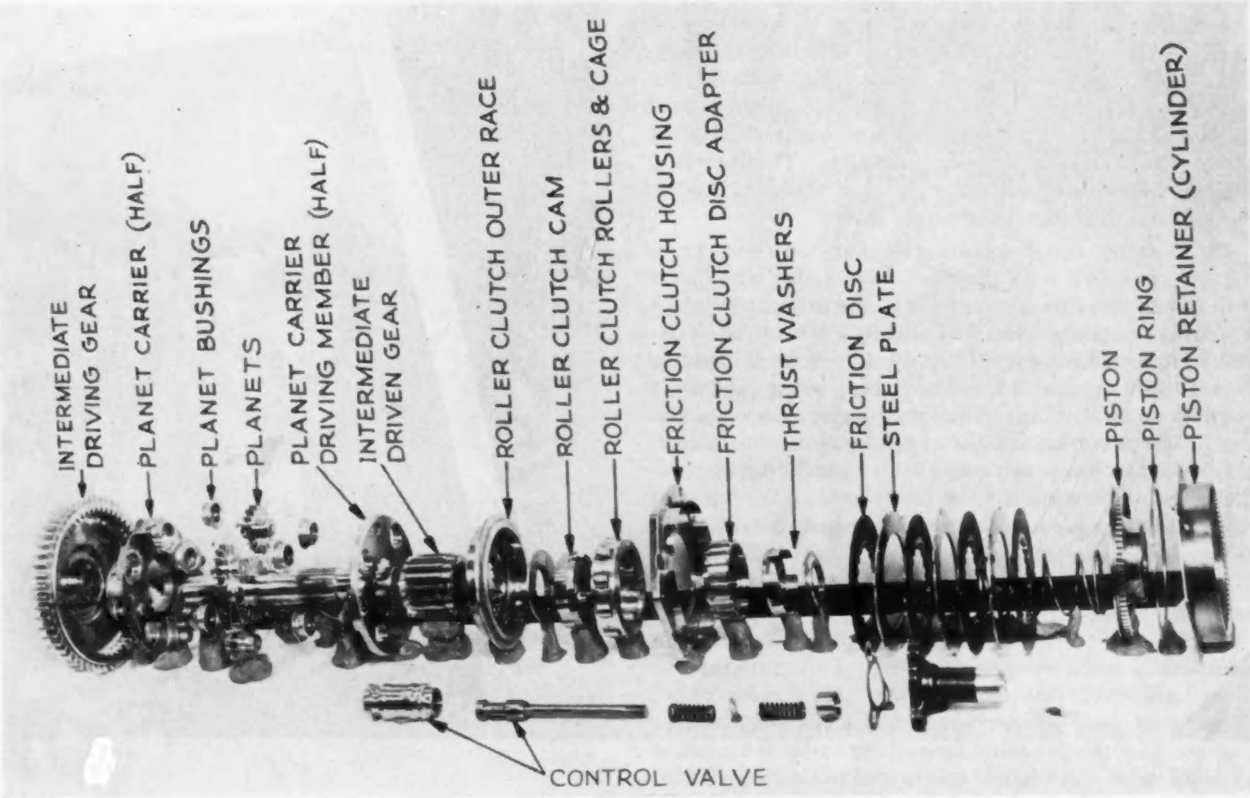
■ Wright Two-Speed Roller Clutch

The 7.14:1 and 10:1 ratios do not meet the requirements of all operators. Some require a combination consisting of a 7:1 ratio for "take-off" at low altitudes and a medium ratio of about 8.5:1 for "take-off" at medium altitudes. They are not interested in high-power cruising at high altitudes. The Wright Aeronautical Corp. can furnish a "close" ratio drive of 7.13:1 and 8.6:1 as well as the more normal ratios of 7.13:1 and 10:1 in a somewhat simpler and lighter-weight type of two-speed drive.

This new drive incorporates a roller clutch in the low ratio. It eliminates the rotating friction clutch in the high ratio and in its place uses a stationary friction clutch in the rear cover in the same space as the low-speed clutch in the old plate type, and a planetary step-up gear in the space formerly occupied by the high-ratio friction clutch.

Fig. 12 shows an exploded assembly of this drive. The large intermediate gear greatly resembles that in the old type clutch. Instead of the splines and the provision for the hydraulic cylinder on the inside, it has a row of internal gear teeth for a planetary gear system. Just to the right of it can be seen the planetary gears which take the place of the old friction clutch. The intermediate shaft has on it a small sun gear for the planetary system instead of the trunnions for the reduction gear pinions in the old type. The small intermediate gear resembles that in the older drive except that it has splines on both ends. On the rear end there is splined an outer race for the roller clutch which resembles in general appearance the internal gear for the old reduction-gear system. In the space formerly occupied by the reduction gear planetary pinions, there is now a roller clutch cam and cage with ten rollers. The high ratio friction clutch element is almost identical to the old type low ratio clutch except that it has four clutch discs instead of three.

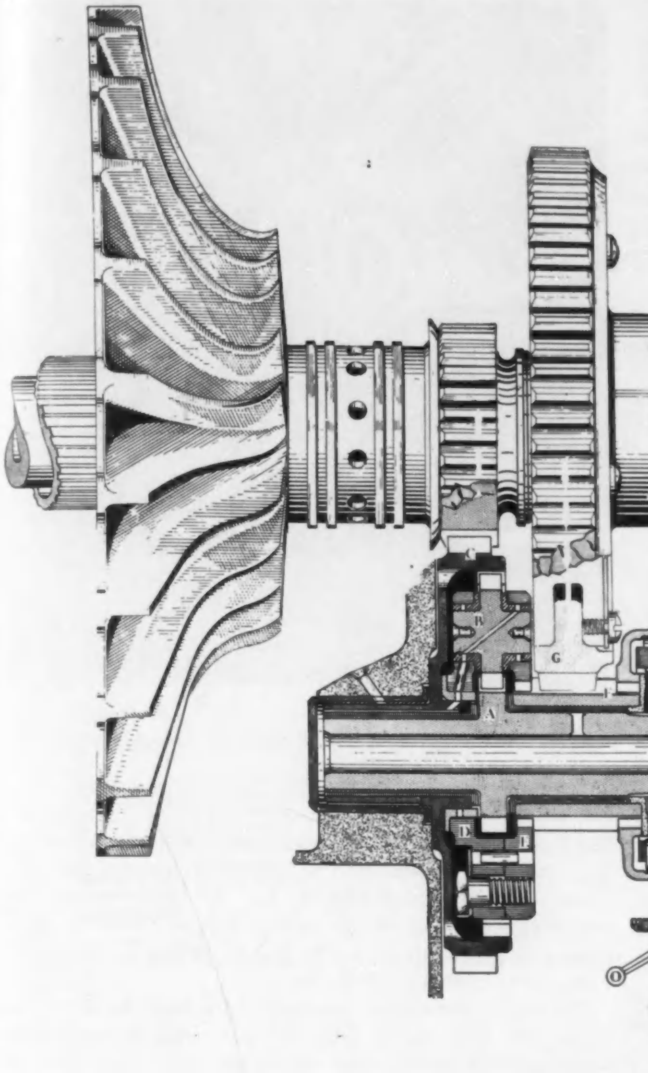
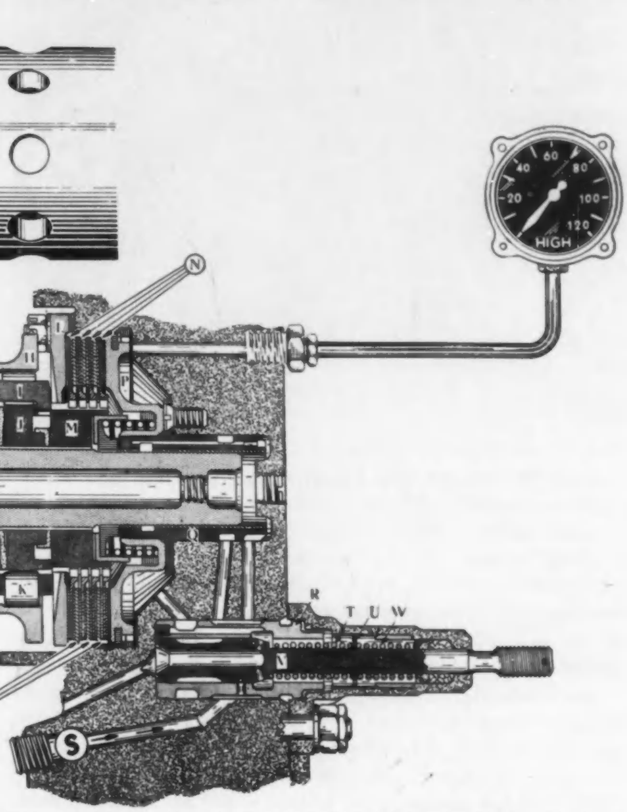
Fig. 13 shows the roller clutch unit in both the engaged



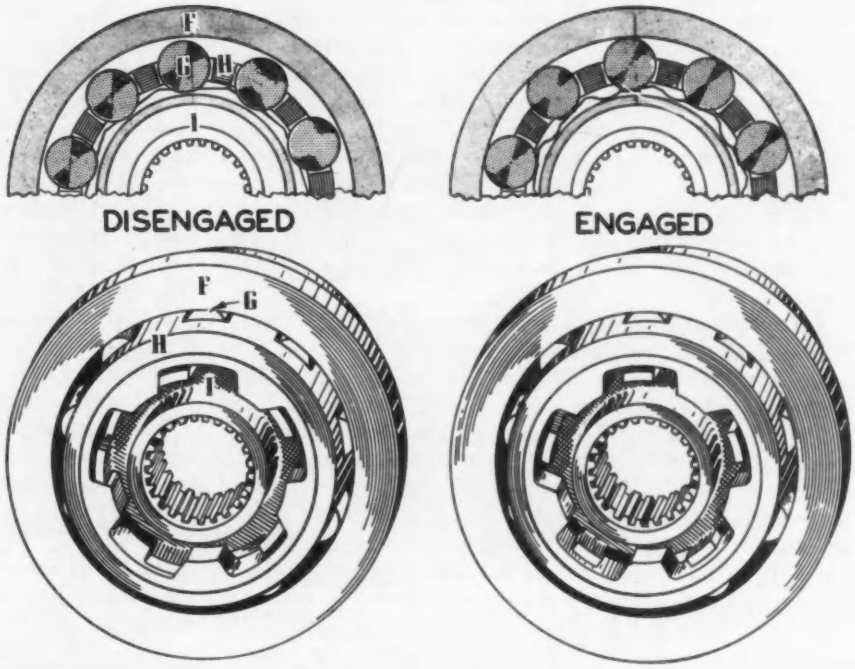
■ Fig. 12 – Wright roller clutch, exploded view

and disengaged positions. When the cam is turned in a counter-clockwise direction, the flats or small inclined planes on this cam cause the rollers to be pushed outward and become firmly wedged between the cam and the outer race, locking the two parts together. When the cam is turned in a clockwise direction, the space between the outer race and the cam flats becomes greater, freeing the rollers and allowing the cam to turn. This unit is identical in principle to the automotive type of free-wheeling unit.

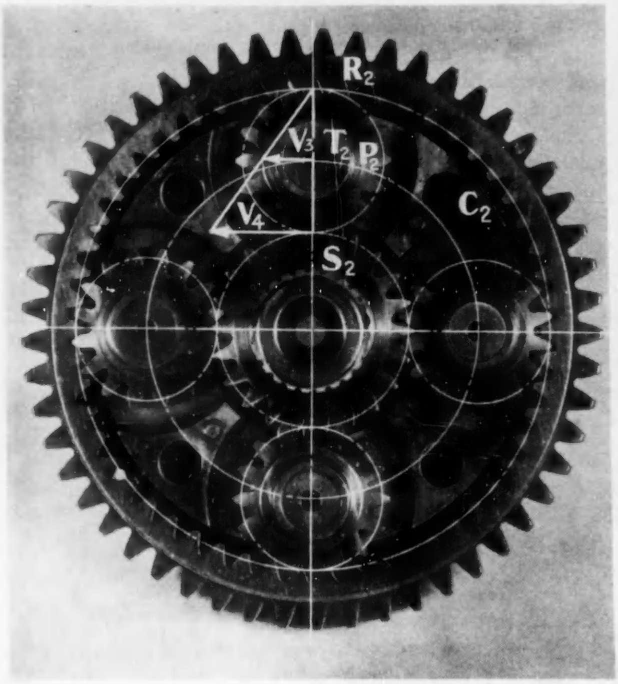
Fig. 14 shows this clutch assembly with the 7.134 and

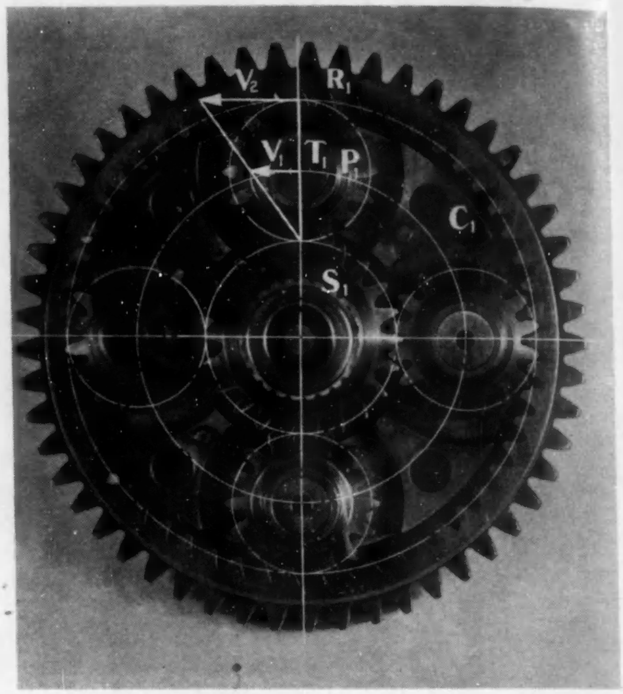
▪ Fig. 14 – Wright roller clutch two-speed drive assembly



▪ Fig. 13 – Roller clutch unit



■ Fig. 15 – Wright roller clutch planetary system, low ratio



■ Fig. 16 – Wright roller clutch planetary system, high ratio

10.04 ratios. When engaged in the high ratio, the tail-shaft gear G drives a small intermediate gear F exactly as in the old type of clutch. When the high-ratio friction clutch, consisting of piston P, friction discs, N, steel plates O, and housing L, is engaged, the intermediate shaft A, with an integral sun gear, is held stationary. The planetary gear carrier D-E, splined to the propeller end of the small intermediate gear F, with its four pinions B then drives the large intermediate gear C at a speed of 1.4 times that of the small gear F with an overall impeller ratio of 10:1.

When engaged in the low ratio, the high-ratio friction clutch is released and the reaction on the sun gear in the planetary system causes the intermediate shaft A to rotate faster than the small intermediate gear F causing the cam J in the roller clutch unit to wedge the 10 rollers K between the flats on the cam and the outer race H of the roller clutch unit. This locks all of the rotating parts in the assembly into the equivalent of one solid compound gear and the drive is then exactly like that of the single-speed supercharger drive shown in Fig. 3. The ratio in low blower is 7.134:1.

An examination of Fig. 15 will show why this gear system locks the roller clutch in the low ratio. Under these conditions the impeller load on the internal gear R_2 tends to hold it stationary. The driving member is the carrier C_2 supporting the pinion with the integral trunnions T_2. The velocity vector V_3 represents the velocity of the trunnion T_2. This tends to impart a velocity V_4 to the sun gear S_2; that is, the direction of the sun gear S_2 is counter-clockwise with respect to the carrier C_2, but the roller clutch will not allow the sun gear to rotate in this direction and it securely locks the sun gear and carrier together. The high-ratio operation is illustrated in Fig. 16. In this case the sun gear S_1 is held stationary by the high-ratio clutch. The velocity vector V_1 represents the movement of the trunnion T_1 supported by the carrier C_1. The pinions P_1 then tend to drive the internal gear R_1 at the velocity represented by V_2. In this diagram the direction of rotation of the sun gear S_1 is clockwise with respect to the carrier C_1. The roller clutch will permit the sun gear to rotate in this direction.

The maximum power transmitted by these drives is 125 hp in low ratio and 190 hp in high ratio, an impressive figure considering the small size of the unit, less than 6 in. in diameter and weighing only 15 lb for the complete drive.

One of the most interesting questions about a two-speed supercharger drive is the time required for engaging the clutches. The double-plate type drive requires 3 sec for shifting from low to high ratio and 0.5 sec for the shift from high to low. The roller-clutch type of drive will shift from low to high ratio in 0.3 sec and from high to low in 0.7 sec.

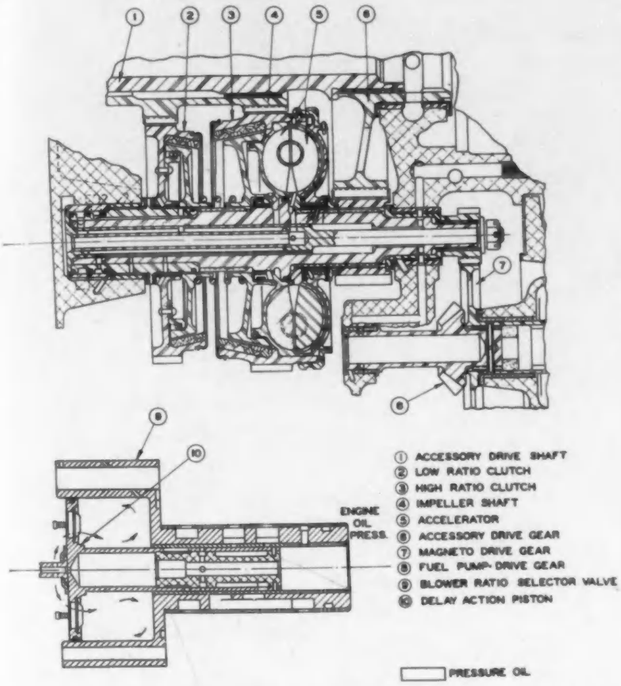
This time was measured by actuating a set of magneto breaker points by an extension of the supercharger drive layshaft projecting through the rear cover. The impulse from the breaker unit was passed through an amplifier and recorded on 35-mm film by a recording oscillograph. A second channel in the oscillograph was used to measure the engine speed by recording on the same film an impulse induced in the circuit by one of the spark-plug leads.

This method of measuring speeds is especially valuable in testing a new design. It will give a precise answer to a number of important questions.

■ Pratt & Whitney Two-Speed Drive

Pratt & Whitney uses two parallel clutch units to drive the supercharger impeller in order to reduce the size of the gears (Fig. 17). Each unit is mounted on an intermediate shaft running at slightly less than three times engine speed. The low ratio and high ratio intermediate

spur gears are coupled to the shaft by rotating hydraulically actuated cone clutches. To decrease the load on the clutches during engagement, a fluid coupling is connected to the high-ratio intermediate gear to accelerate the driven members of the cone clutches to a speed between the high and low ratio speeds. After the cone clutches are engaged,



■ Fig. 17 – Pratt & Whitney selector valve and clutches in neutral

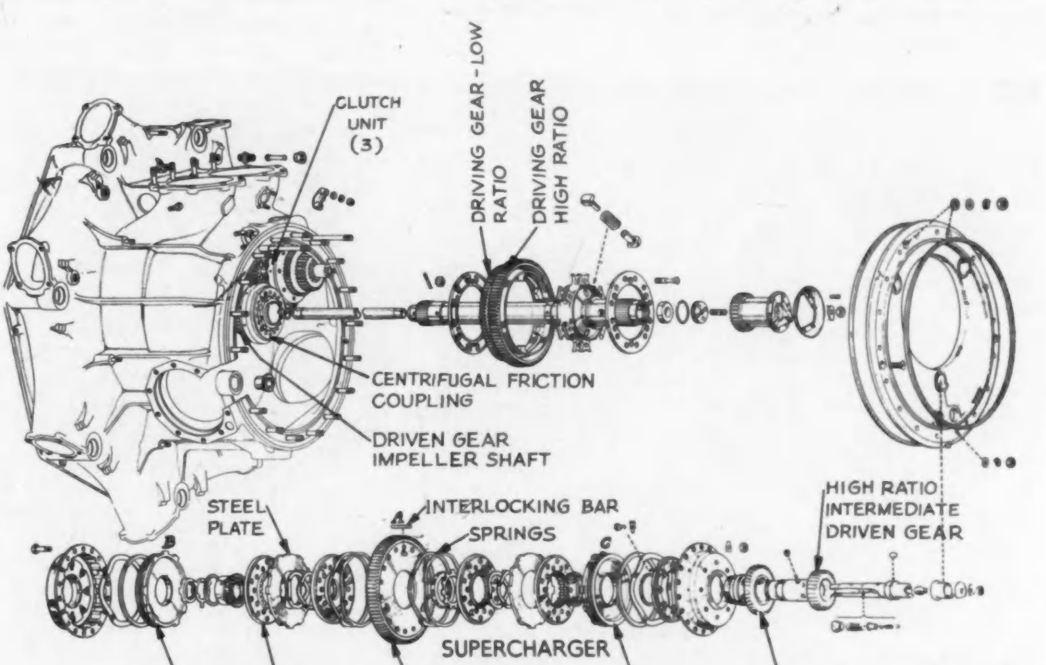
the control valve cuts off the supply of oil to the fluid coupling.

A dashpot in the selector valve unit prevents engagement of the cone clutches until after the accelerator has been engaged a predetermined period of time. This time is fixed by an orifice in a dashpot piston. For the return from the high or low ratio positions to neutral position, two single-direction oil relief valves permit the valve to be moved rapidly without hindrance by the orifice. A thermostat is incorporated in the selector valve to maintain a uniform oil temperature and consistent engaging speeds.

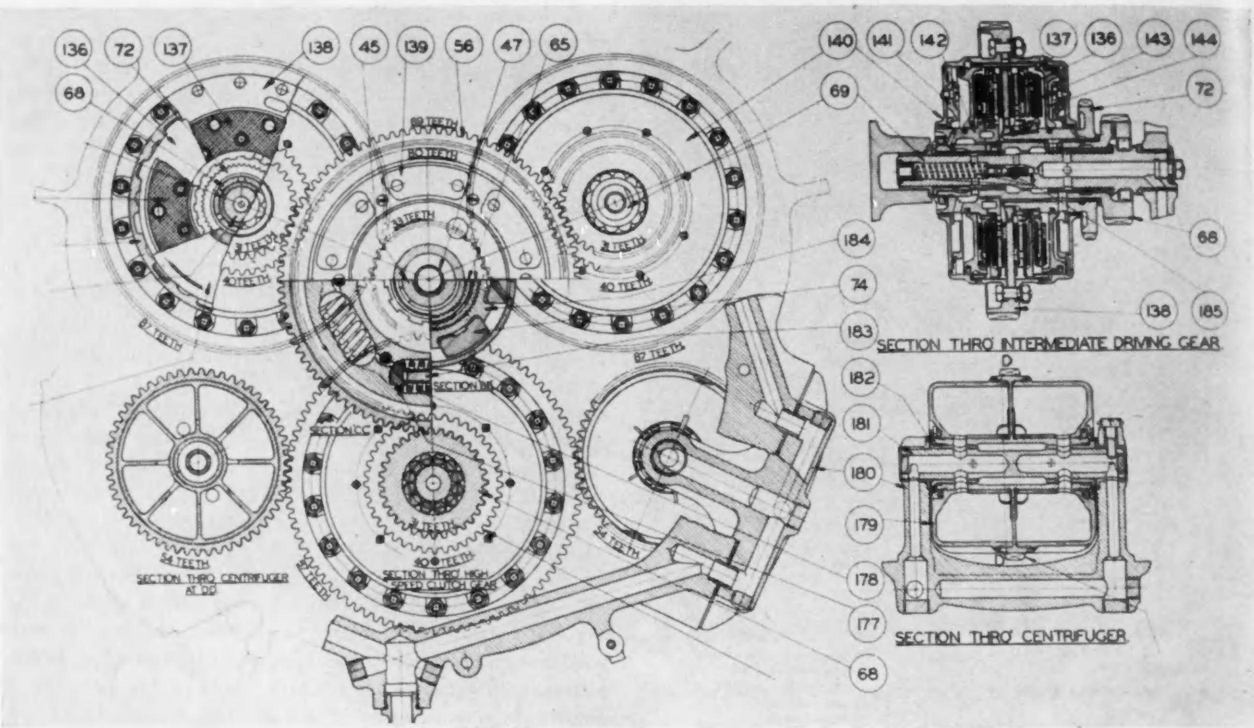
■ Bristol Two-Speed Supercharger Drive

The Bristol two-speed drive is similar to the Wright design. An exploded view is shown in Fig. 18. It consists of three identical double-acting clutch units equally disposed about the central driving tailshaft and the concentric impeller shaft. Each unit contains two driven gears meshing with two spring-mounted driving gears on the tailshaft. Each of these driven gears can be coupled to the large intermediate driving gear by a clutch. The assembly is shown in Fig. 19. When the clutch (Part Nos. 136, 137, 143, 144) on gear 72 is engaged the drive is in the low ratio, 5.27:1. When the clutch on gear 68 is engaged the drive is in the high ratio, 7.56:1. The actuating oil pressure is admitted to these clutches through the hollow stationary layshaft. The double check valve provides a means of lubricating the clutch bearings without adding a special passage for the lubricant. These clutches are prevented from engaging simultaneously by a set of interlocking pins shown in Fig. 18.

The clutches in both ratios rotate with the common housing. Two centrifuges are provided to remove the sludge from the oil before it enters the clutch actuating cylinders. These centrifuges are driven from the lower of



■ Fig. 18 – Bristol two-speed supercharger drive, exploded



■ Fig. 19 – Bristol Hercules two-speed supercharger drive assembly

the three intermediate driving gears. In addition to preventing sludging of the clutches, these centrifuges probably clean the complete engine lubricating system. The centrifuges are readily removable from the rear section for cleaning at regular intervals.

■ Rolls-Royce "Merlin" Supercharger Drive

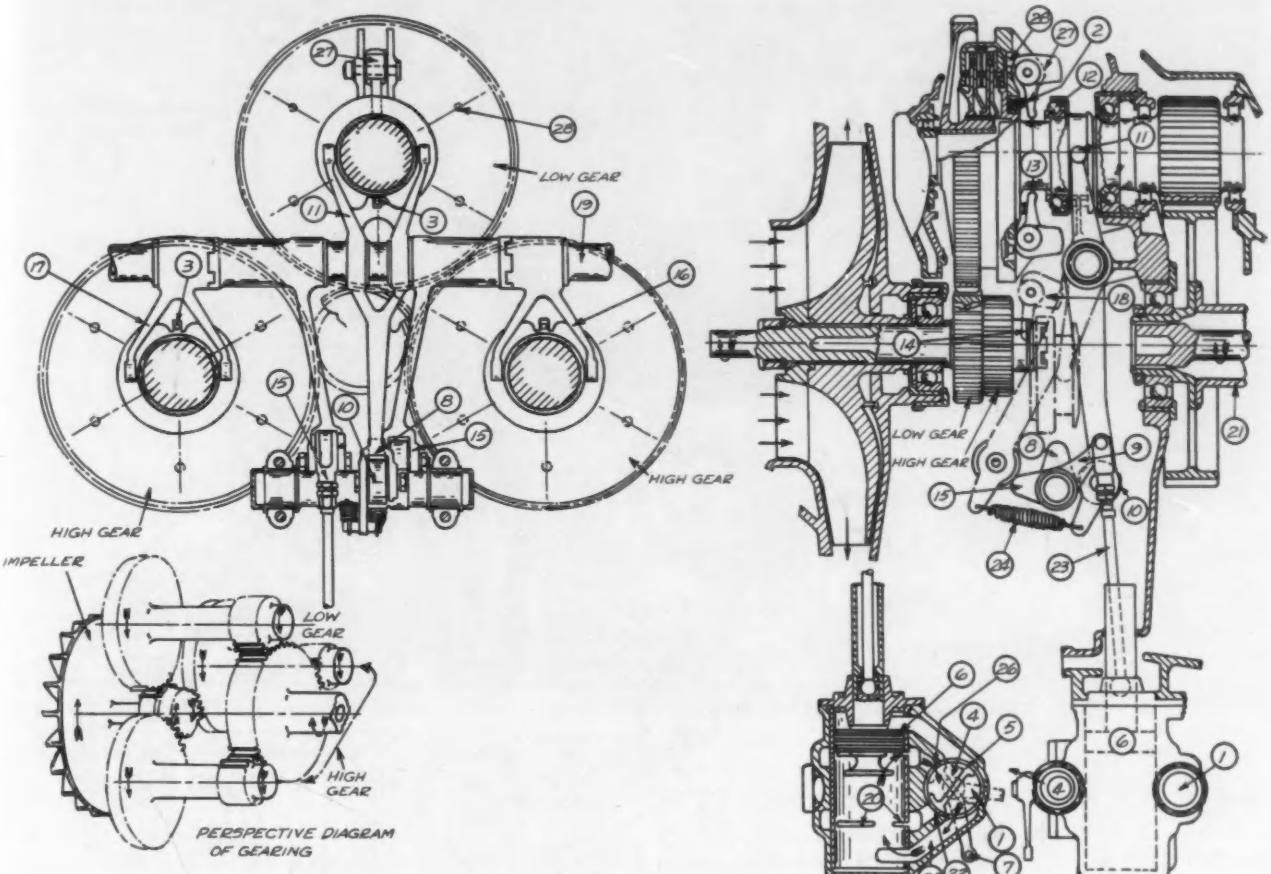
The Rolls-Royce "Merlin" two-speed drive is shown in Fig. 20. This drive is furnished in ratios of 6.39:1 in low and 8.75:1 high or 8.15:1 low and 9.49:1 high.

It has three clutch units equally disposed about the central driving gear on shaft (21) and the "low" and "high" gears on the impeller shaft. One of these, the upper one, is used for low ratio only, and the two lower ones are used for high ratio only. These clutches are of the semi-centrifugal type similar to those in automotive use. The actuating force is furnished by the springs (2), and the centrifugal weights (27). The clutch is disengaged by the thrust bearing (12) operated by the forked levers (11), for the low ratio and (16) and (17) for the high ratio. The levers are held in the disengaged position by cam (8) for low ratio, and (15) for high ratio. The camshaft is actuated by a hydraulic piston (6) through link (23) and lever (9). The operating pressure for the piston is supplied by the engine scavenging pump and controlled by valve (5). Except when the piston is moving from one extreme to the other, part of the scavenging oil is circulating through the cylinder, entering through passage (25), or its equivalent on the other end, and leaving through ports (26).

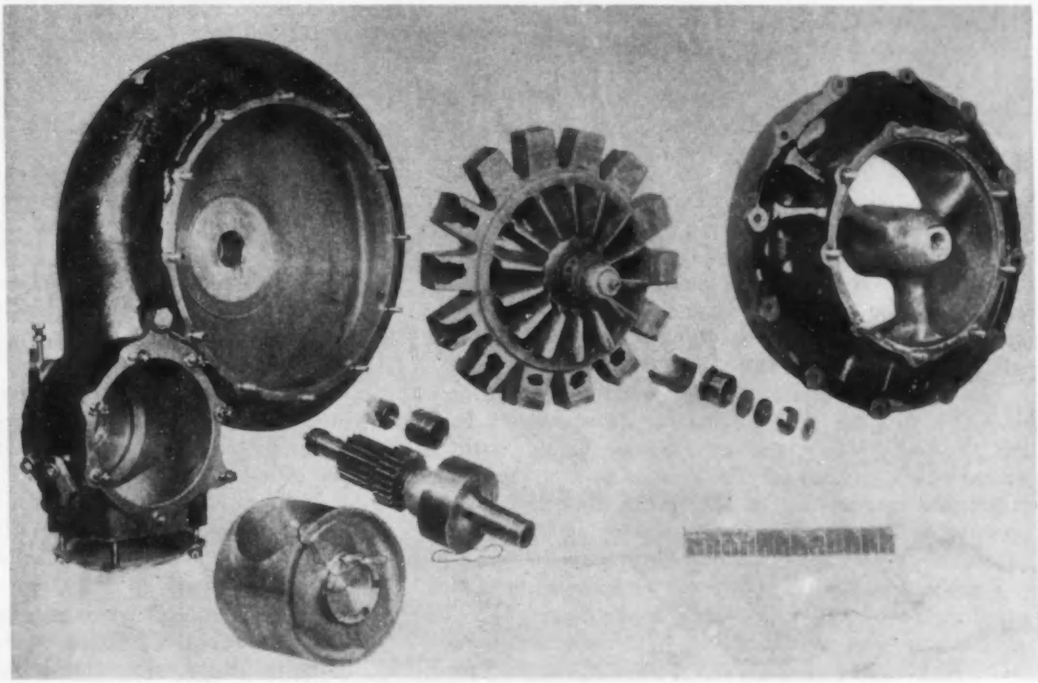
■ Junkers Jumo 211 Two-Speed Drive

The Jumo 211A supercharger is shown in Fig. 21. The barrel throttle on the outlet and the spout-type impeller are interesting. This impeller was damaged by rubbing on the housing because of a failure of the impeller bearings. The oil lines from the oil tank to the engine were shot away but, as the main bearings were just starting to fail, it was apparent that the impeller bearings had failed before the fight started. The cause of the failure was due to an inadequate oil supply to the bearings. The lubrication to these bearings was improved in later models.

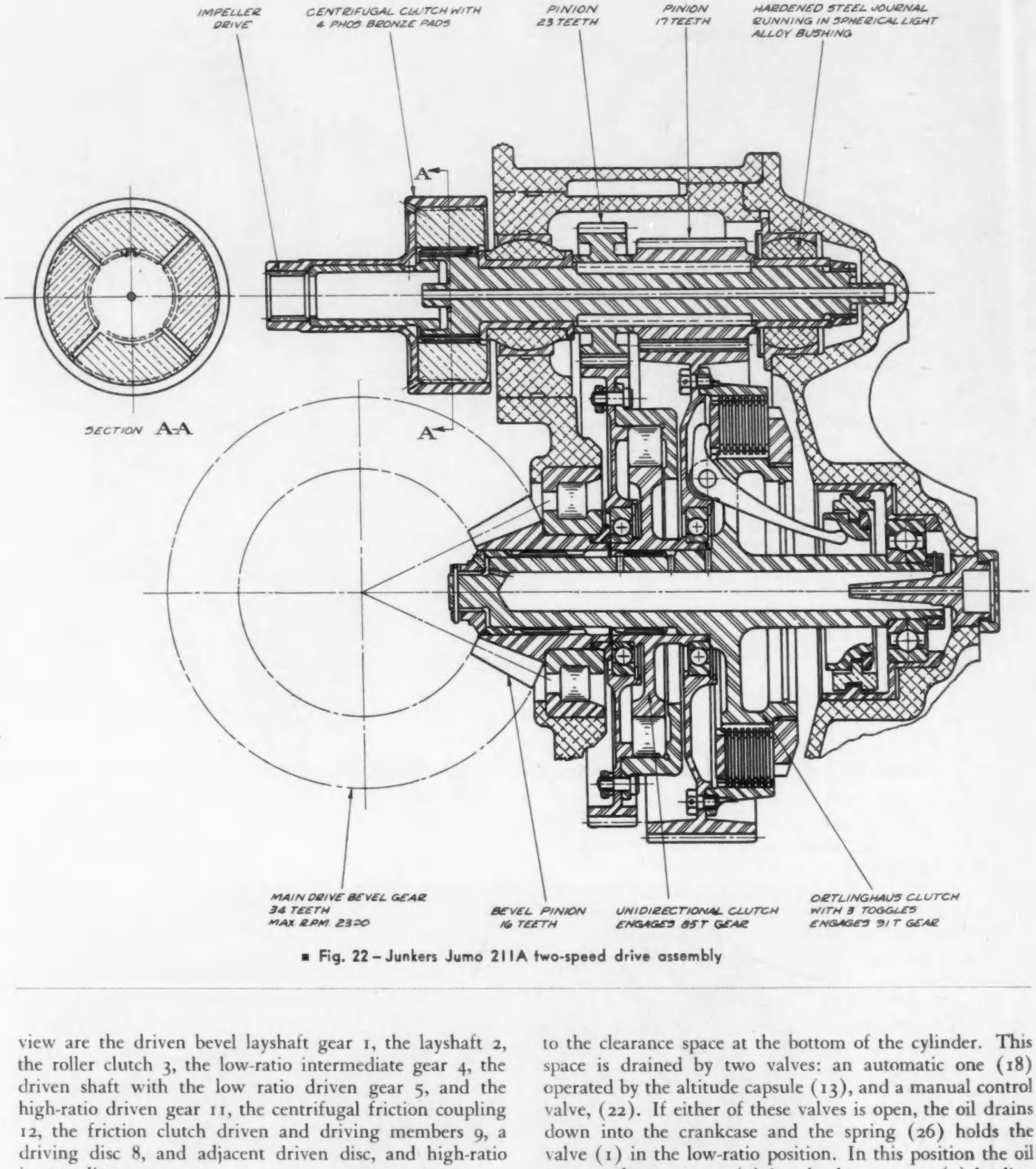
Fig. 22 is an assembly of the two-speed drive. The large bevel gear, on the crankshaft, drives the small bevel on the layshaft. In the low ratio (7.95:1) the layshaft is connected to the 85-tooth spur gear by a roller clutch. The 85-tooth gear meshes with the 23-tooth pinion on the final driveshaft. This shaft is coupled to the impeller shaft by a centrifugal friction coupling. In the high ratio the larger intermediate gear is coupled to the driving layshaft by a friction clutch of the Ortlinghaus type, commonly used in industrial applications in Germany. The large intermediate gear drives the 17-tooth pinion at 11.375 times the engine speed. The distinguishing feature of the Ortlinghaus clutch is the method of applying pressure to the clutch plates. A sleeve, tapered on the inside, is moved axially over three levers contacting the movable clutch plate. In this case, the axial movement of the sleeve is caused by a rotary motion of another concentric sleeve having helical splines on the outside diameter. This part is shown attached to lever 6A in Fig. 23. Other major parts in this



■ Fig. 20 – Rolls-Royce Merlin two-speed supercharger drive assembly



■ Fig. 21 – Junkers Jumo 211A supercharger



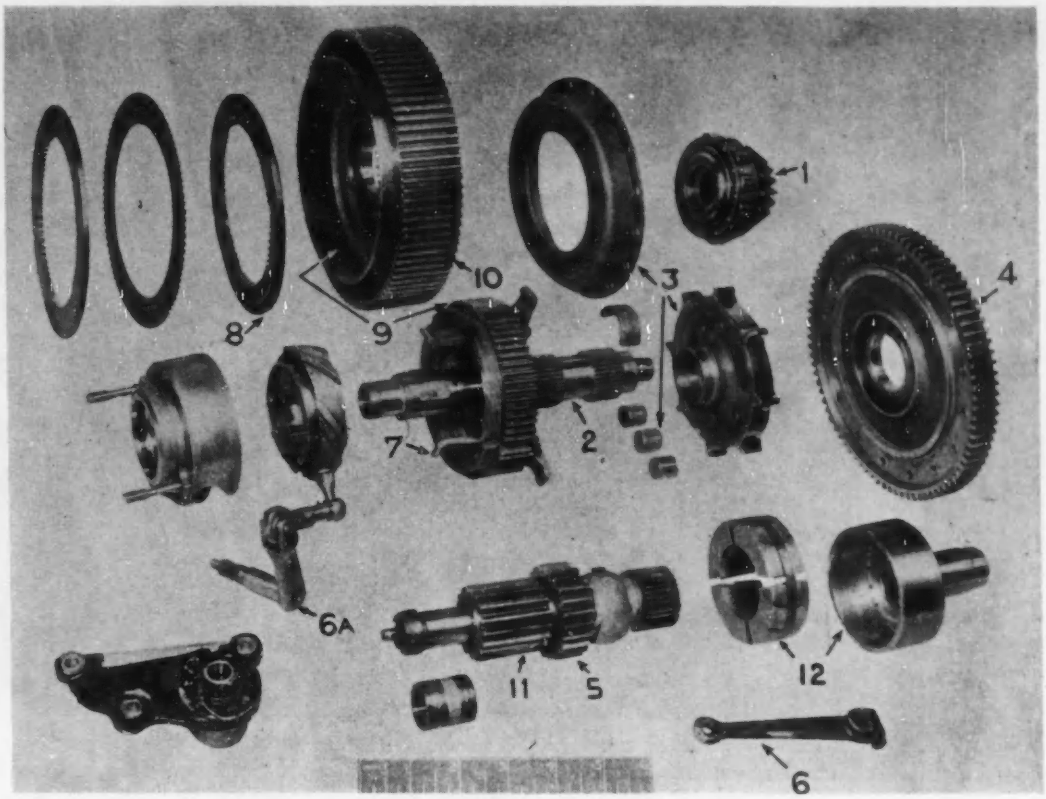
■ **Fig. 22 – Junkers Jumo 211A two-speed drive assembly**

view are the driven bevel layshaft gear 1, the layshaft 2, the roller clutch 3, the low-ratio intermediate gear 4, the driven shaft with the low ratio driven gear 5, and the high-ratio driven gear 11, the centrifugal friction coupling 12, the friction clutch driven and driving members 9, a driving disc 8, and adjacent driven disc, and high-ratio intermediate gear 10.

The control valve for the Junkers two-speed drive is shown in Fig. 24. The mechanism incorporates a barometric capsule which prevents operation in high ratio at altitudes below 10,000 ft. The manual control valve (22) determines the position of the spring and hydraulically operated valve (1). This valve supplies oil to the servo vane that actuates the two-speed shifting mechanism.

There is a constant supply of oil pressure to the central annular groove in valve (1) through oil passage (5). This oil bleeds through the radial holes in the valve and around the spiral passage formed by the threads on the plug (10), to the clearance space at the bottom of the cylinder. This space is drained by two valves: an automatic one (18) operated by the altitude capsule (13), and a manual control valve, (22). If either of these valves is open, the oil drains down into the crankcase and the spring (26) holds the valve (1) in the low-ratio position. In this position the oil pressure from passage (5) is valved to passage (7) leading to the low-ratio side of the servo vane. But, if both the manual and barometrically operated valves are closed, the oil bleeding through the spiral passage builds up pressure in the cylinder and forces the valve (1) to the upper position as shown in the diagram. In this position the oil from passage (5) is valved to passage (6) and the high ratio side of the servo vane. The manual control valve allows the oil pressure built up in the control valve cylinder to be applied to the layshaft bearings, augmenting their lubrication. Detents prevent hunting of the valve.

If the manual control valve is shifted to the high-ratio



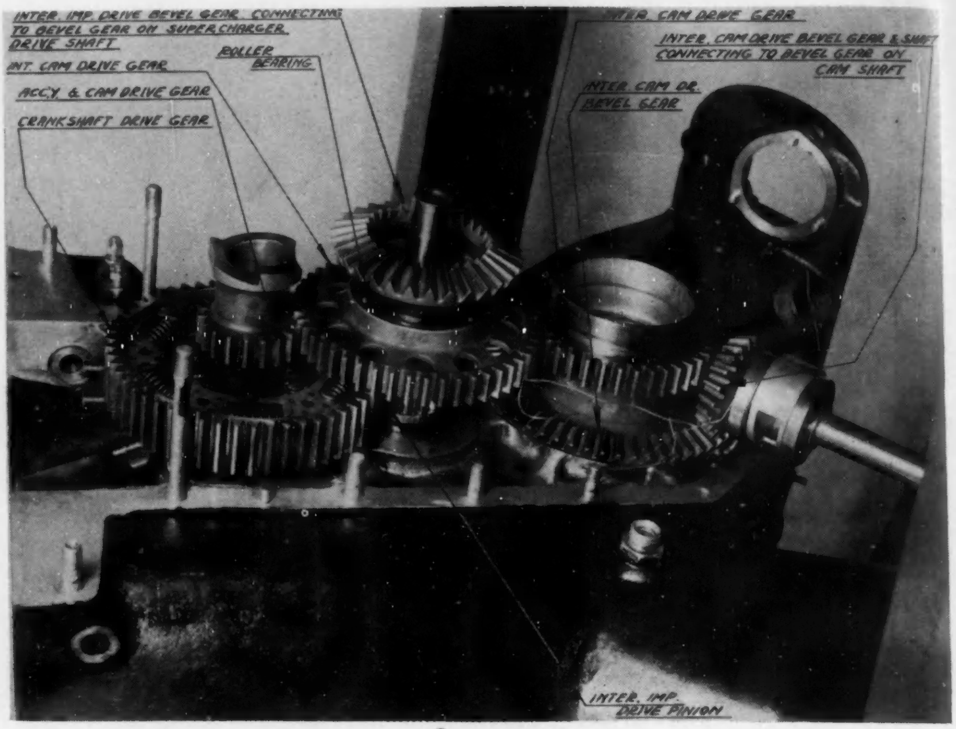
■ Fig. 23—Junkers Jumo 211A two-speed parts

position at altitudes below 10,000 ft, the oil in the cylinder drains out through passage (19) past valve (18) and through passage (23) to the crankcase, and the main valve (1) remains in the low-ratio position.

■ DB-601A Variable-Speed Drive

The Mercedes-Benz DB-601A engine utilizes a fluid coupling in the supercharger impeller shaft to vary the speed of the impeller. The primary step-up gears are

■ Fig. 24—Junkers Jumo 211A two-speed control valve



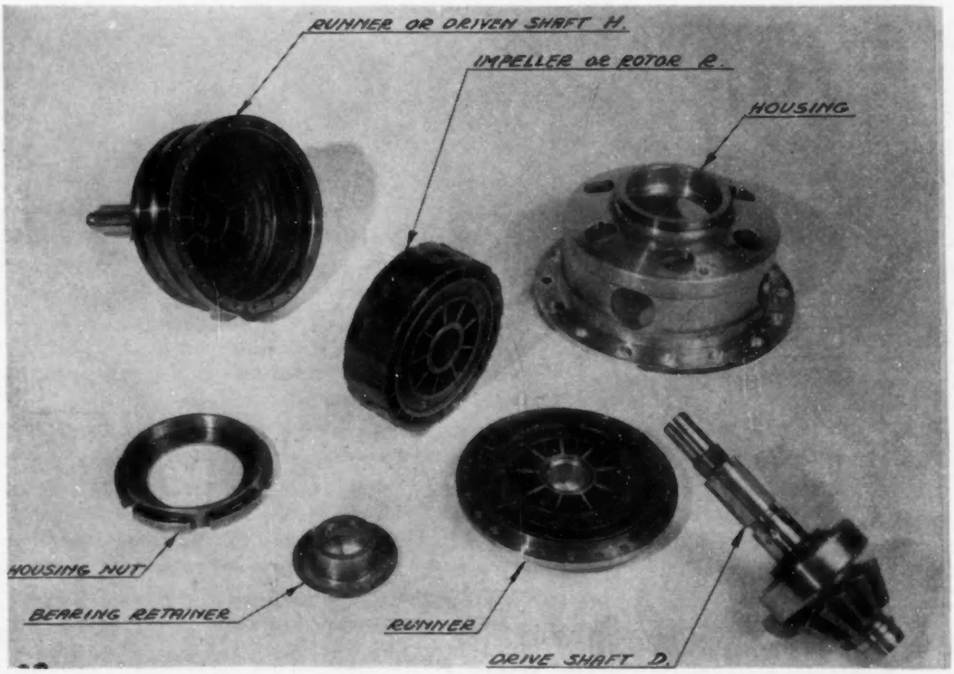
■ Fig. 25 – Mercedes-Benz supercharger drive gears

shown in Fig. 25. The large spring coupled gear on the crankshaft drives the pinion on the layshaft at 4.15 times engine speed. The bevel on the layshaft drives a smaller bevel (not shown) on the final drive shaft with an overall ratio of 10.38:1. A fluid coupling between the final drive gear and the impeller reduces this ratio to something between 7.3 and 10.2:1, depending on load, engine rpm, altitude, and oil temperature.

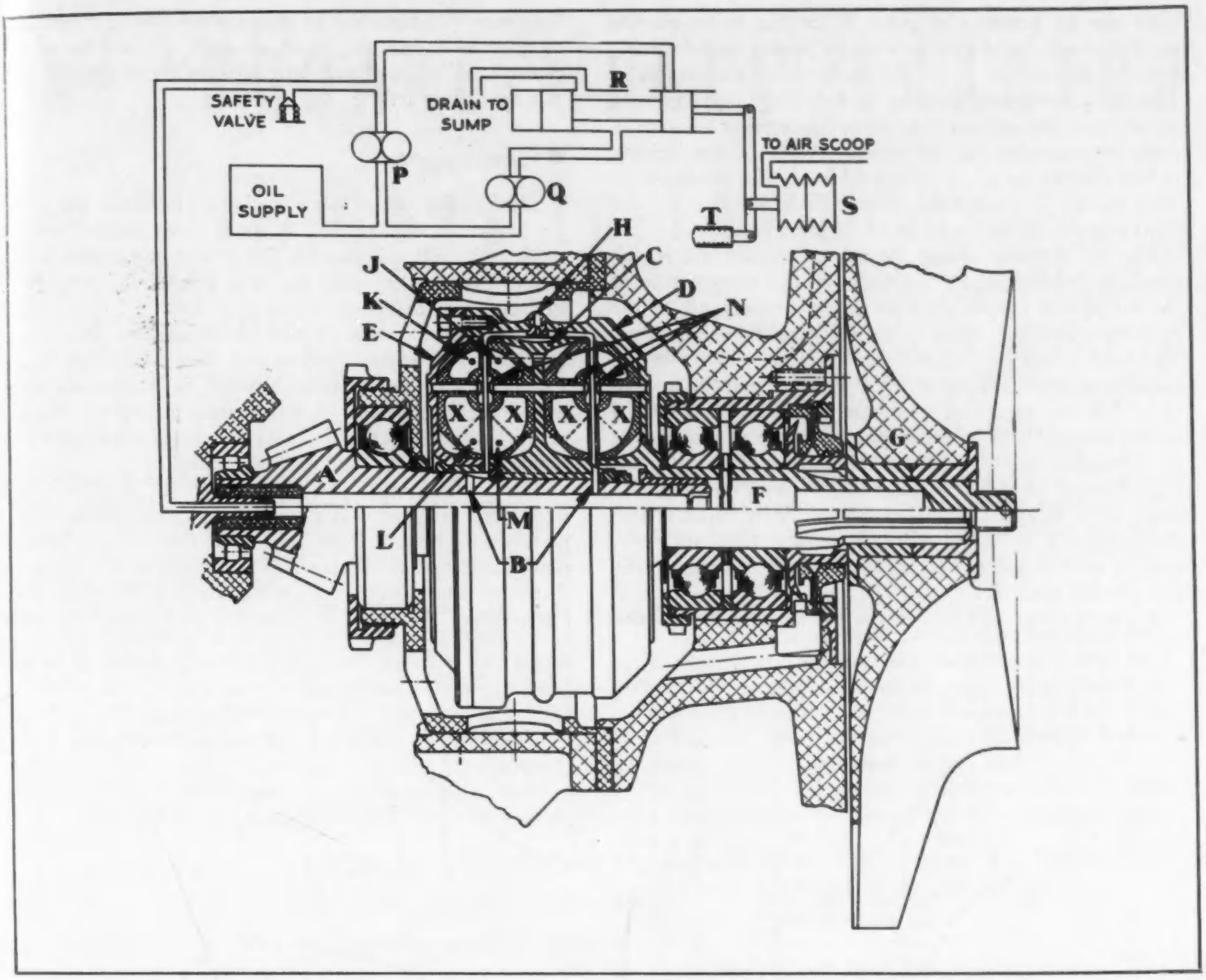
The parts of the fluid coupling are illustrated in Fig. 26. A cross-section of the fluid coupling in combination with a diagram of the complete system is given in Fig. 27. The final driveshaft *A* has splined on it the driving runner *C* of the fluid coupling. The driven member is the housing *D* and its cover *E*. Two each of the die-cast parts, containing 18 fluid cells each, are pressed into both the driving and driven members. Four of the annular rings *N* are riveted in place to provide a core about which the oil circulates. These rings are supported radially by notches in the vanes that form the walls of the fluid cells.

The operating principle is identical to the fluid flywheels now being produced in the automotive field.

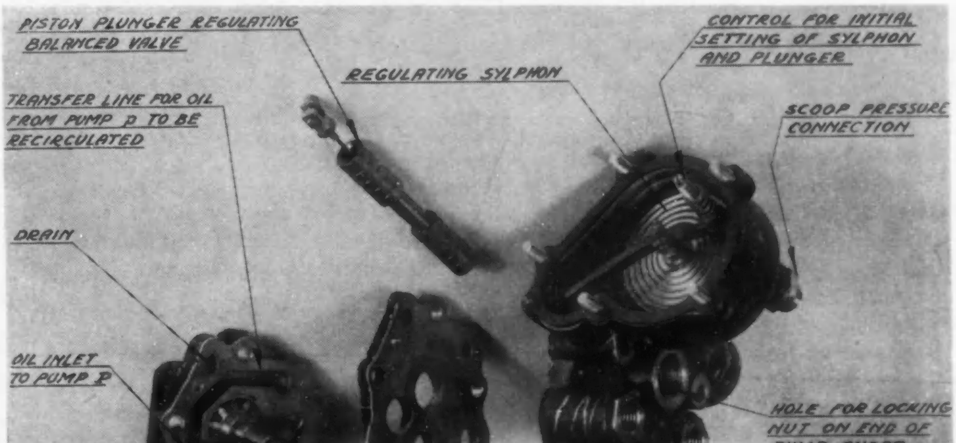
Even under the most favorable conditions, there is some slip in the coupling. Therefore, at point *J* in the driving member, the oil pressure due to centrifugal force is greater



■ Fig. 26 – Mercedes-Benz DB-601A hydraulic coupling parts



■ Fig. 27 – Mercedes-Benz DB-601A variable-speed hydraulic supercharger drive and control system



■ Fig. 28 – DB-601A oil pump and automatic altitude control unit

than the oil pressure at point K in the slower-moving driven member. Oil then flows from point J in the driving member to point K in the driven member, pushing oil at point L in the driven member back into the driving member at point M, and continues to circulate about the core N from one member to the other. While in the driving runner, the oil picks up considerable angular momentum. This energy is transferred to the driven member by the impact of the oil on the vanes of the fluid cells.

The slip of this coupling, for any given load and engine speed, is determined by the quantity and temperature of the oil in the coupling. Two pumps supply oil to the coupling. Pump P operates at full capacity at all times. At rated power and speed the coupling will drive the impeller at a ratio of about 7.3:1 with oil from this pump only. For higher ratios, additional oil is supplied by the second pump Q. A balanced plunger valve R actuated by an aneroid capsule S regulates this flow, allowing the full capacity of the pump Q to enter the coupling for the maximum ratio of 10.2:1, for operation at high altitude, and diminishing it for lower ratios, to zero at 7.3:1, for operation at low altitudes. Fig. 28 illustrates the oil-pump unit and altitude control.

A jet H (Fig. 27) allows the oil to flow out of the coupling continuously, permitting the oil level, and hence the ratio, to be controlled by regulation of the rate of oil flow into the coupling. Another important function of this flow is to cool the coupling.

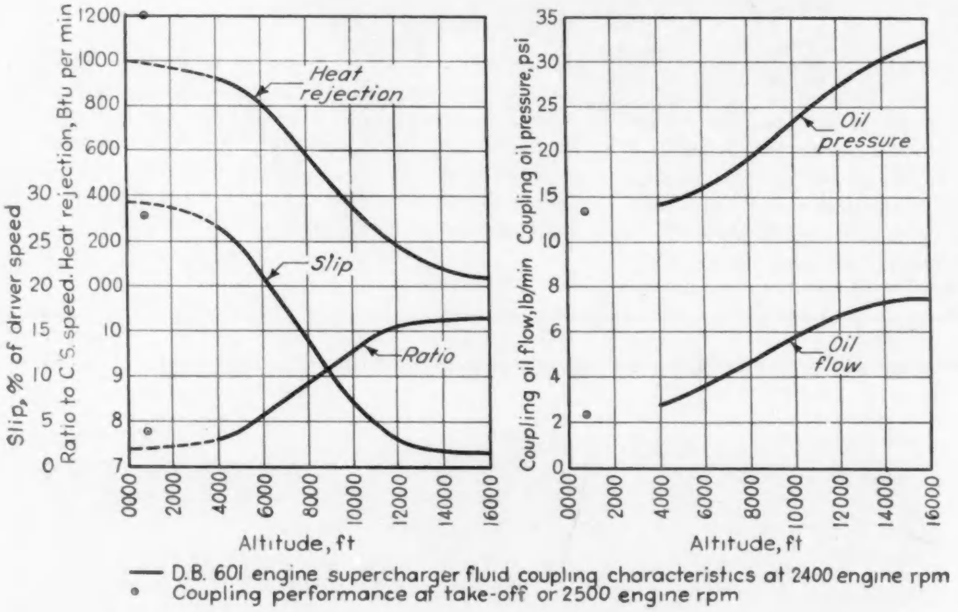
Probably the greatest disadvantage of this drive is the large amount of heat wasted in the churning of the oil in the coupling. The curves in Fig. 29 give the characteristics of this drive. At "take-off" power the heat rejection to the oil approaches 1000 Btu per min. As the slip decreases with altitude, the heat rejection diminishes rapidly to a negligible value. This factor would not be serious in a pursuit-plane installation as the ship would usually "take-off" with a cold engine and rapidly climb to an altitude requiring operation in the high ratio minimum slip condition. The high heat rejection to the oil while climbing would aid in warming the oil system. However, the performance of a transport or patrol plane, flying at altitudes of 5000 ft or less, might be seriously affected by an oil-cooler installation of sufficient capacity for an engine with this type of drive.

Conclusions

The DB-601 design has stimulated interest in this country in the fluid coupling. A great many studies will be made of it, and it is possible that it will be adapted to our requirements. Certainly, we have benefited greatly by a knowledge of its design.

With the exception of the Mercedes-Benz fluid drive (neglecting oil coolers, and so on), the construction of the European two-speed drives appears to be heavier than most American drives. A conspicuous example is the use of thick riveted friction faces on the clutch discs instead of the cemented facing used in this country.

The moderate altitude performance of the two-speed single-stage engine is not impressive compared with the performance required for this war. Greater emphasis is now placed on modifications of two-speed drives, adapting them for use with two-stage geared superchargers with intercoolers. An increasing number of installations with the now firmly established exhaust-gas driven turbo-supercharger will also tend to decrease the interest in two-speed superchargers.



■ Fig. 29 – DB-601A hydraulic coupling characteristics

Acknowledgment

I am indebted to the Bristol Aeroplane Co., Rolls-Royce Ltd., and the Bendix Aviation Corp. for information on the British designs, to the British Air Ministry and the Royal Aircraft Establishment for information on the German designs and to Messrs. Roland Chilton, Arthur Nutt, and S. T. Robinson for obtaining this information; to Messrs. Vincent Moore, S. T. Robinson and Raymond Young for their criticisms; and to Messrs. P. Bancel, A. L. Beall, K. Campbell, J. H. Critchley, Paul Evans, H. D. Jackes, L. Mraz, E. F. Pierce, C. M. Shay, and J. Talbert for other information and assistance.

The "SQUARE-WHEEL" TRACTOR Goes to Town

by ROBERT MAYNE and H. W. DELZELL
The B. F. Goodrich Co.

SPEED and mobility of thousands of combat units of the expanding United States mechanized army have been increased tremendously by the use of rubber tracks on crawler tractors, the authors declare.

Placed on army tanks, scout cars, anti-aircraft and anti-tank gun carriers, they contend that the rubber tracks have given our mechanized forces a flexibility of operation never before achieved. This is because the rubber track permits much higher speeds, allows the vehicle to use any type of highway without damaging it, and operates as well in varied terrain, it is explained. The steel track did not permit this easy operation on the highways without damage to them. The rubber track has greatly expanded the zone of operations of any unit so equipped.

Lessons gained from the use of rubber tracks on combat vehicles will prove invaluable in the adaptation of the track to commercial uses after the emergency is over, the paper declares.

The day when rubber tracks applied to the crawler-tractor became practicable is as important in the progress story of the crawler as the first application of a pneumatic tire to a rolling wheel, the authors point out.

Many of the most desirable characteristics of the rubber tire and steel track are combined when the rubber track is used on crawlers, these including low rolling resistance, freedom of the crawler from noise and vibration, operation at high speeds, high efficiency and traction, the paper states.

THE crawler tractor of today is a far cry from the clanging, lumbering machine from which it sprang only a few decades ago. Through rapid engineering advances, it has become a dependable piece of equipment, neatly proportioned and easily handled.

In spite of all the improvements, the wheel tractor man has never become thoroughly reconciled with the crawler and it is with a mixed feeling of envy and scorn that he refers to it as a "square-wheel" tractor. With envy, because of its long, flat ground contact; with scorn, because of the noise and vibration still attending its operation.

But now, through the use of rubber, the so-called "square-wheel" tractor can travel with the quiet, effortless speed which, previously, had been associated only with the round wheel.

Perhaps some of you saw, a few years ago, a movie news release in which a farmer and his wife went riding to town on a crawler tractor. They drove through the downtown section of a busy city with the same apparent ease and speed as did the rest of the traffic.

Of course, it was all a movie man's idea of good publicity, and it is not likely that the farmer will make a regular practice of going shopping in crawler tractors, even when equipped with rubber tracks, but he will surely be able to take to the highway for quick and smooth transportation when the need requires.

It seems that the day "a square-wheel tractor went to town," even though it was only for publicity purposes, marked an important step in the progress of the "square wheel." It marked the successful application of rubber to the crawler, an achievement which, in the history of the development of the crawler, may prove of comparable importance to the successful application of rubber to the wheel.

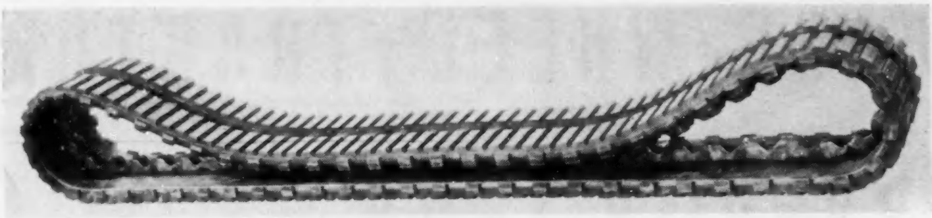
Our company started the development work on a rubber track about ten years ago. The work was carried out initially under the direction of C. W. Leguillon, manager of our Machine Development Department, and it is because of his foresight and perseverance in the face of considerable difficulties that the undertaking was successful.

The steel track has been improved considerably by the use of rubber bushings which reduce friction in the joints and by the application of rubber blocks to permit operation on the highway. But the B. F. Goodrich track is more than a mere application of rubber to the steel track; it is a complete departure from past practice.

In its essential features, this track consists of a rubber band reinforced longitudinally by steel cables to which a number of steel cross members are attached at regular intervals. These cross members carry the guides and, in the case of sprocket-driven tracks, contact the teeth of the sprockets. The assembly forms a substantially inextensible structure of great strength and flexibility. The following discussion is limited to tracks of this general type.

A number of variations of the basic design have been developed, each suited to particular applications. From

[This paper was presented at the National Tractor Meeting of the Society, Milwaukee, Wis., Sept. 25, 1941.]



▪ Fig. 1 – 12-in. continuous band track – sprocket drive

these variations, we shall select three designs to illustrate the range of possibilities of the track:

Fig. 1 illustrates one of these designs. It is designated as a "continuous band track" because it is molded integrally into a single continuous piece. The comparative light weight and low rolling resistance of this track make it particularly suitable for high-speed operation.

▪ Fig. 2 – Continuous band track mounted on Army half-track scout car

Fig. 2 shows the track mounted on a vehicle that will be recognized as the U. S. Army half-track scout car. The tracks and the front-wheel drive give remarkable cross-country mobility to this vehicle which is also capable of sustained high speed on the highway. As you probably know, half-track scout cars are being produced in large quantities for our mechanized forces and will constitute one of their main offensive units. The track is driven by means of a sprocket contacting metal bosses on the cross members. The guides are interlocked in a sort of fish-scale fashion so that the roller flanges are led from one guide to the next with little chance of climbing. The method has proved very effective in all types of operation.

This arrangement is necessary because the rubber track, unlike a steel articulated track, has little inherent torsional or lateral rigidity. This characteristic is very desirable in many respects, but it increases considerably the problem of satisfactory guiding. In all cases where a constant high track tension cannot be maintained, the foregoing method of guiding has been found necessary.

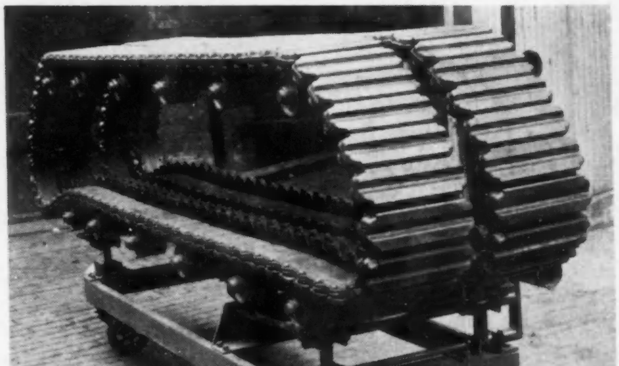
Steel grousers can be applied to the track whenever necessary to increase traction. A comparatively low grouser is used to hold vibration down to a minimum when the vehicle has to be operated over a hard-surface road with the grousers attached.

Fig. 3 shows a so-called band block track which differs from the previously discussed band track in that its tread is made up of a number of separate blocks bolted to the cross members. This design is heavier and perhaps not as well suited to the high speed possible with the band track. It has the advantage, however, of providing greater traction and of making possible the replacement of the tread blocks when worn. It is felt that this latter feature will bring about an important saving in track cost per mile.

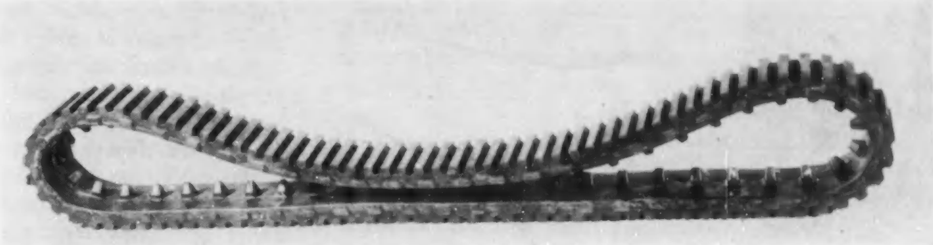
This block type of track also has the advantage of being easily adapted to a variety of service conditions. It will always be possible to use a tread surface which is best suited to a particular service by installing blocks of the proper design. A number of block designs, incorporating various compromises between long life and traction, have been made available. For some operations, as for example on ice, it will be possible to replace the rubber blocks by steel grousers to provide maximum traction.

The two types of track already discussed were designed to meet difficult operating conditions with regard to speed and severity of service. The need of an inexpensive track for use on a farm or industrial tractor has long been recognized. Fig. 4 shows an experimental track intended for that need.

As you may see, the design has been reduced to the



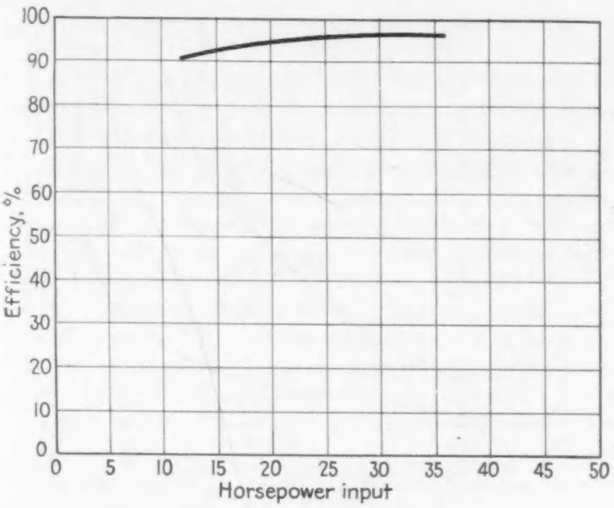
▪ Fig. 3 – Band block track with tread made up of separate blocks bolted to the cross members



■ Fig. 4 – Experimental band track for use on farm or industrial tractor – friction drive

simplest possible expression. The guides are integral with the cross members. The more complicated interlocking guides used in the other two type tracks have been abandoned in favor of the simpler design. This is made possible by operating the track under tension to offset the torsional and lateral flexibility.

The drive of this track is interesting. In one form it depends upon herringbone ribs on the drive wheel matching with grooves on the inside of the track. In another form it depends solely upon friction. Naturally, the whole success of this type drive is based upon the possibility of



■ Fig. 5 – Indoor dynamometer test – Goodrich rubber track Type T-10 – speed, 30 mph; initial tension, 1000 lb

maintaining sufficient tension on the track to prevent slippage under all conditions. The drive would be impractical on an undercarriage such as that of the half-track scout car, where the track tension varies with the deflection of the bogie springs.

In actual test it was found that no driving slippage took place when wet clay was fed on the face of the driving wheels and the tracks were slipped on dry concrete and, under these conditions, the best possible traction prevailed between the tracks and the ground and the worst between the track and the drive.

Friction-drive rubber tracks, using fabric reinforcement, have been tried in the past but were found to be unsatisfactory because they lacked the strength necessary for sufficient tension. The steel cable reinforcement of the Goodrich Rubber track overcomes this limitation.

This track design is still experimental, but has already shown definite promise. If proved successful in further tests, it will have the advantage of eliminating the sprocket.

Other types of tracks are being tested or are in the process of development. Space does not permit an exhaustive discussion of these various types, but it is hoped that the three designs just outlined will have given some indication of the possibilities of the rubber track.

These designs have common advantages, the most important of which is perhaps the saving in power.

A considerable amount of power is lost in a steel track as the result of friction in the joints, friction between roller flanges and the guides, and because of excessive vibration.

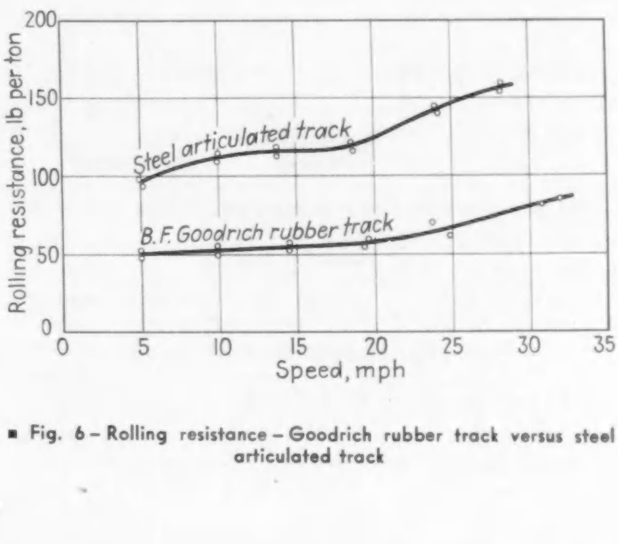
It can be said that, for all practical purposes, the loss of power due to friction in the joints is eliminated entirely by the Goodrich rubber track.

Fig. 5 bears out this statement. It shows an efficiency curve of the track being operated as a belt on a dynamometer. It will be noted that, at a speed of 30 mph with an initial tension of 1000 lb and a load of 35 hp, the efficiency is 97%. A part of the 3% losses could no doubt be accounted for as windage.

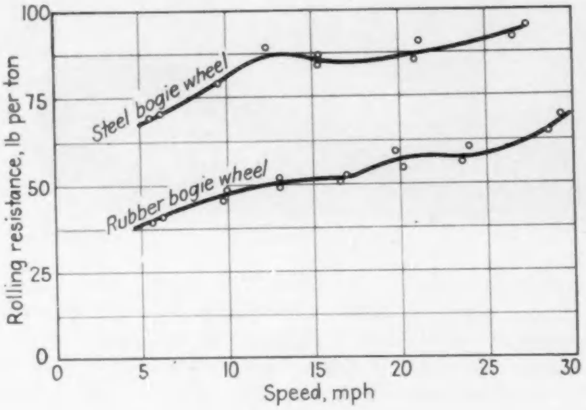
The rubber track is guided much as a steel track by guides contacting roller flanges, and losses from this source must, therefore, be approximately the same for the two tracks. It is felt, however, that the sprocket losses are lower in the rubber track because of a greater accuracy of pitch length. This accuracy is maintained throughout the life of the track because the cables are substantially inextensible, whereas wear in the joints of a steel track has the effect of increasing the pitch.

The vibration losses are reduced greatly by the rubber track, particularly, as pointed out later, when rubber bogie wheels are used.

Fig. 6 shows a comparison of the rolling resistance between a half-track vehicle equipped with a continuous band track and the same vehicle equipped with an articulated track. This articulated track had been designed to



■ Fig. 6 – Rolling resistance – Goodrich rubber track versus steel articulated track



■ Fig. 7 – Goodrich rubber track rolling resistance – steel wheel versus rubber bogie wheel

the end of reducing friction to a minimum. It had a pitch of only 2 in. and lubricated bronze bushings. It was equipped with rubber blocks so that the road surfaces of the two tracks were approximately the same.

The curves show that the saving of power made possible by the continuous band track, at 30 mph, is in the vicinity of 40%.

It will be noted that the saving increases with speed. This is as should be expected because the friction losses in the joints increase with the centrifugal force which, in turn, is proportional to the square of the speed and, as just pointed out, these losses are practically non-existent in the rubber track.

Although the power saving is greatest at high speed, it will be noted that a substantial saving is indicated even at the lowest speed.

■ Rubber Wheels Save Power

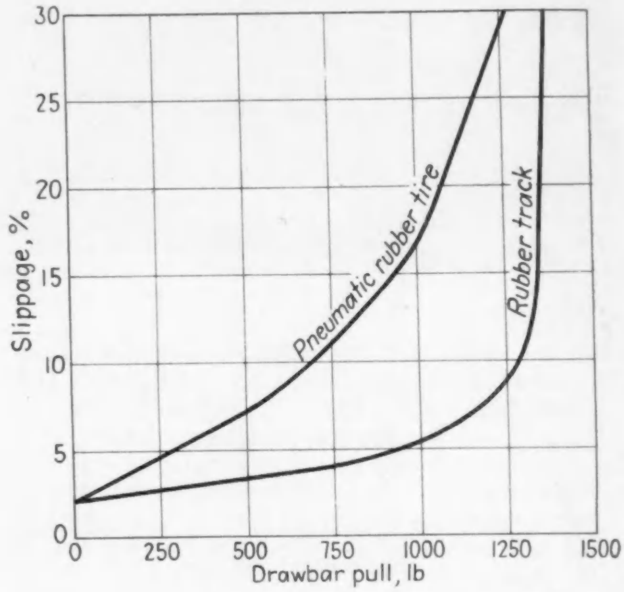
Fig. 7 reveals that a surprisingly large power saving is made possible by the use of rubber instead of steel bogie wheels. Rubber bogie wheels have the effect of reducing vibrations beyond what had already been accomplished by the use of a rubber track in place of a steel track. The saving in power indicated by the curves results from this additional smoothing out of the operation. It brings out clearly the large power losses resulting from vibrations and the effectiveness of rubber in reducing these losses.

Table 1 indicates the comparative rolling resistance at 3 mph of a crawler-tractor equipped with a steel track, a crawler-tractor equipped with a rubber track, and a rubber-tired tractor. It will be noted that the resistance to traction of the rubber track lies between that of the tire and the steel track. It will be noted also that, as the ground becomes softer, the resistance to traction of the track approaches that of the tire. It is felt that, in still softer ground, the rubber track would have considerable advantage over the tire.

The comparison of the traction between a rubber track and a pneumatic tire is interesting. Fig. 8 shows such a comparison. The curves were taken with a tractor equipped with rubber tracks and one equipped with pneumatic tires, the two tractors being of approximately equal weight and power.

TABLE 1 – COMPARATIVE ROLLING RESISTANCE AT 3 MPH

Rubber-Tired Tractor	Rubber-Track Tractor	Steel-Track Tractor
3160 lb total weight	3040 lb total weight	7070 lb total weight
	Hard, Dry Soil	
32 lb per ton	66 lb per ton	127 lb per ton
	Dry Clay – Loosened to a Depth of 4 In.	
95 lb per ton	131 lb per ton	170 lb per ton



■ Fig. 8 – Rubber track versus pneumatic tire – soil condition, loose, dry clay – total weight of rubber track, 3040 lb; of pneumatic tire, 3160 lb (rear 2681) with 12 psi air pressure

It will be noted that the curve of traction against slip in the case of the rubber track is very flat. A drawbar pull very near the maximum is reached by the rubber track with comparatively small slip. You will note, for instance, that, for a slip of 10%, the drawbar pull developed by the track is about 85% greater than that developed by the tire; for a slip of 7%, it is 135% greater.

The ability of the rubber track to develop traction with comparatively small slip is invaluable. When a tire is operated at a 17% slip, the same proportion of the available drawbar power is lost. If a track can develop the same torque at one-third this slip, the loss is thereby reduced in the same ratio.

The practical drawbar pull at which a tractor is operated is determined not so much by the maximum pull that it can develop but by a maximum slip beyond which it is not economical to operate. The rubber track, therefore,

offers the possibility of a higher efficiency at lower slip, combined with greater drawbar pull, for the same weight of tractor and same power.

It must be pointed out that flotation was not a consideration in the conditions under which the foregoing curves were taken. The ground was dry Illinois clay which had been loosened to a depth of only 4 in. In a soft plowed field the track would have shown much greater advantage over the tire.

With regard to wear, our data are far from complete. On an army vehicle the service varies so much that no average figure can be given. On the highway at high speed the track will naturally wear at a comparatively fast rate. In sand, there is practically no wear. Fig. 9 illustrates a rubber track operated on a farm tractor for about 2000 miles. You will note that the wear is hardly noticeable. When the test was discontinued after about twice this mileage, the tracks did not indicate any more wear than shown in this photograph. From this and other tests, we feel confident that the wear of a track will present no serious problem in average commercial applications.

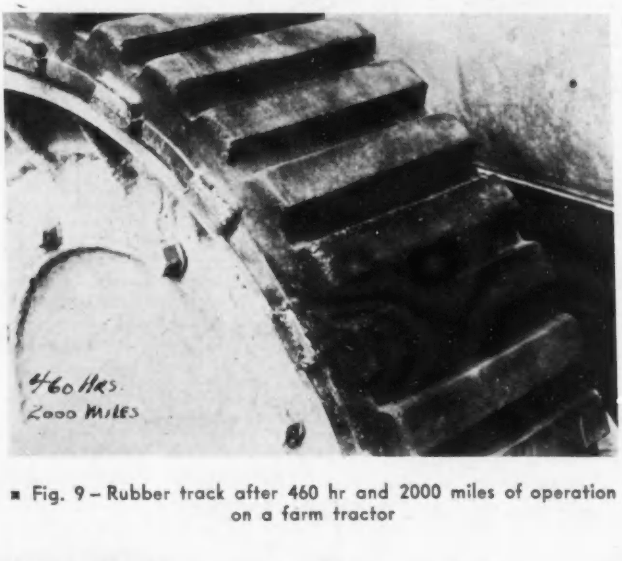
With its low rolling resistance, freedom from noise and vibration, high efficiency, high traction, suitability for high-speed operation, the B. F. Goodrich Co. rubber track combines many of the most desirable characteristics of both the tire and the steel track.

Just as these characteristics have made it an outstanding contribution to the tools of war, they can make it an equally important contribution to the tools of peace.

Under the present emergency, much experience is being gained in the manufacture and use of rubber tracks. Because of this experience, we are now better able to undertake the problem of adapting the rubber track to a farm or an industrial tractor.

The application of the rubber track to commercial fields is not a job that the rubber manufacturer can do alone. It must be a joint undertaking with the vehicle manufacturer. A successful application depends as much upon a properly designed undercarriage as upon a satisfactory track.

On the basis of past accomplishments, it is our conviction that rubber can do for the crawler what it has already done for the wheel.



■ Fig. 9 – Rubber track after 460 hr and 2000 miles of operation on a farm tractor

Requirements for Tomorrow's Light Plane

ENGINEERING of small airplanes has progressed tremendously in the last two years. We now know the necessity of doing away with parasite obstructions even in a small ship. We know the spin-proof, stall-proof characteristics that can be had and, before we tool up too far for production on already outmoded types of dangerous vision and poor flying characteristics, the industry must look itself in the eye and face facts.

It is time for the "puddle-jumper" to grow up. The period has come when it must adopt the new ideas of the transport, the military plane, and the pursuit ship to give a consistent performance with its horsepower if it is to hold its place as an industry. There is no continuous market in this country for a plane with a 70-mph cruising speed where the Mississippi valley has 55-mph winds three months in the year.

When this is done, however, the private-owner plane can not have the requirements for finesse of piloting that is available on airliners or in military equipment. These planes must be designed to be safe in the hands of even 10-hr pilots.

My definition of a safe airplane is one that will *get into the smallest field, over the highest trees, on to the roughest ground, in the shortest distance, with the poorest pilot, without any trouble*. It is, of course, an advantage that the plane should be able to take out of the same field safely over the same trees.

Much progress is being made in the study of such planes. *The greatest deficiency of the modern small plane is vision.*

Motor stoppage, which used to be the prime danger in the air, is now greatly secondary to the factor of collision in the air. In none of these high-wing planes can you see the sky above and back of you to see if some other plane is coming in to land on your wing. Nor can you see forward and below to be sure you are not landing on somebody else!

When you want to make a right-hand turn, the right-hand wing goes down in front of your vision so you are turning to the right without being able to see where you are going – like a motor car would be if you had no side window. With this type of vision, of course, these planes cannot be used in a military way nor in formation and the Army cannot be blamed for not being enthusiastic about them in quantity for military use.

All airline passenger ships coming into crowded airports keep an eagle eye out for these small planes which they know are blind in most directions and are a potential danger to other ships in the air. This is a bald statement but it is a statement of fact and must be faced.

Another method has been to build low-wing airplanes. The low wing in the large plane with considerable inertia and sluggishness of action is a practicable set-up. With the big span, enough dihedral can be put into the wing so that it is well balanced and reasonably controllable.

When you take the same idea for a small airplane, however, the problem is completely different. When we design a small airplane, we cannot design a small man to fly it. The cabin has to be the same size with the same ceiling room as in any other airplane.

Excerpts from the paper: "The Small Plane and Its Powerplant," by William B. Stout, Stout Engineering Laboratories, Inc., presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Oct. 30, 1941.

AIRCRAFT CARBURETOR AIRSCOOPS on FUEL-AIR

THE design of a carburetor airscoop involves:

1. Location of the intake opening to receive full air-speed ram and exert minimum drag (which involves knowledge and experiment as to the *external air flow* around the airplane); and

2. Conducting the intake air to enter the carburetor with uniform pressure across its stream with minimum turbulence, and minimum loss of impact pressure (which involves the study of *internal air flow*).

In general, the two main objectives, maximum ram and minimum disturbance of carburetion, go hand in hand and have the same structural requirements.

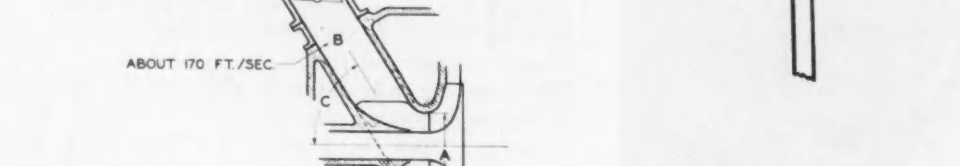
An analysis is given of the principles of fuel-air metering, by measuring the air velocity differential through air orifices of fixed size, and transmitting this differential, corrected for variations of air density, to secure corresponding fuel flow differential through selected fuel orifices. A description is then given of the departures from steady

IN our recent intensive efforts to carry airplane performance to higher speeds and altitudes, we have encountered many complex problems in apparently simple development of previously satisfactory practice. Ignition, cooling, and fuel vapor control are only a few instances. Similarly, the design of an aircraft carburetor airscoop would appear to offer only elementary questions of design. How to locate the opening where it will receive full air-speed ram, and how to fair it in with the cowling structure, would seem to lie well within current knowledge; actually there are indications that many present designs could be improved. Likewise, many of our ideas as to the effect of airscoops upon carburetion have been derived from the past when carburetors were non-automatic, requiring continuous readjustment by the pilot as soon as the airplane left the ground for changes of air pressure, temperature, and ram, and any accompanying disturbances in the scoop duct system could usually, though not always, be taken care of by the same manual adjustment.

The automatic carburetor has now become indispensable for military use and has also proved its worth in commercial service. Rather peculiarly, however, the precedent set with the older non-automatic carburetors of making all tests and carburetor settings on the engine test stand on the ground has, up to this time, been maintained with the automatic carburetors. We now have experience to indicate that the maximum airplane performance cannot certainly be obtained when the carburetor setting is derived from ground tests alone, and that we will have to gain some further knowledge and develop new instrumentation and equipment before we can fully reproduce flight conditions on the ground so far as the carburetor is concerned. Until that time, we must be prepared occasionally to modify our ground-test conclusions and settings by flight check.

This paper deals particularly with some of these flight

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., October 30, 1941.]



■ Fig. 1 – Complete air-intake system, showing discrepancies between internal and external air velocities at rated power at sea level

and Their Effect
METERING in Flight

full-stream air flow encountered in flight service, and the manner in which these affect air-speed metering.

Methods are given for detecting, measuring, and curing disturbances from these sources.

Recommendations also are given for improved warm and cold air control, for protection against ice formation in the intake system, by eliminating the variable ram differential between cold and warm air flow which exists with most current scoop designs.

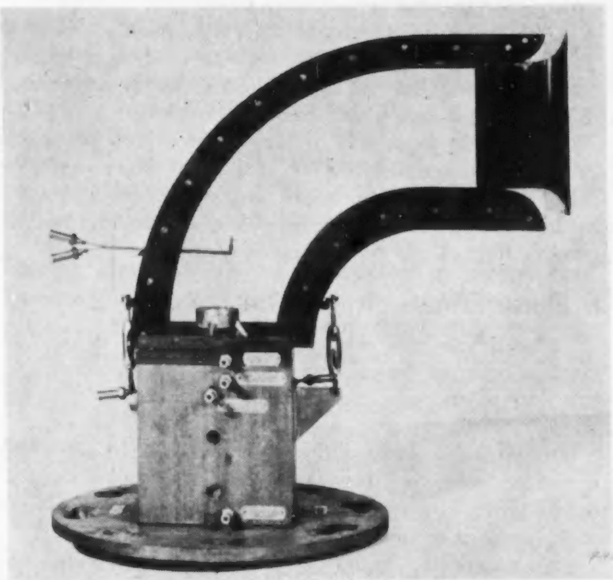
In conclusion, the need is emphasized for freer cooperation between the airplane, engine, and carburetor engineers; and for recognition in our procurement procedure of the need for preliminary check tests in actual flight, on prototype airplanes, of the design and characteristics of not only the scoop and carburetor, but also all other engine accessories whose functioning may be affected by differences between flight and ground-test conditions.

by FRANK C. MOCK
Bendix Products Division, Bendix Aviation Corp.

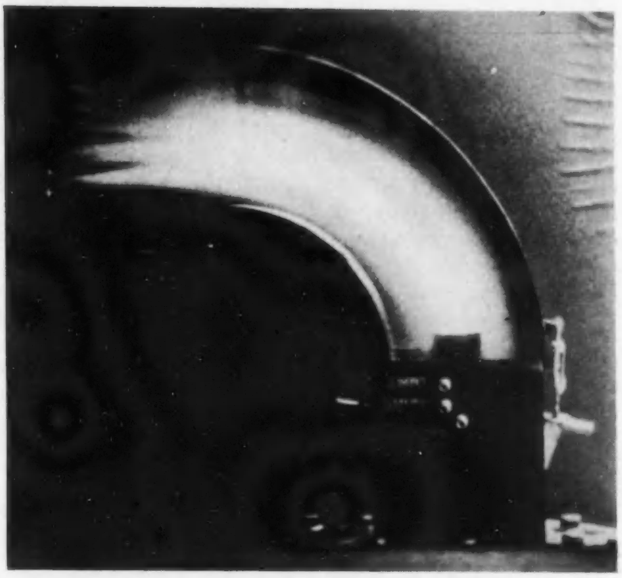
conditions which have not heretofore been taken into consideration in ground test. Some of our work is incomplete, particularly the instrumentation reports, and on such account this presentation is perhaps premature. It is given now, however, in the hope that it may secure immediate attention, and perhaps more effective attack, upon problems which definitely affect the performance of our airplanes.

■ Continuity of Whole Air-Intake System

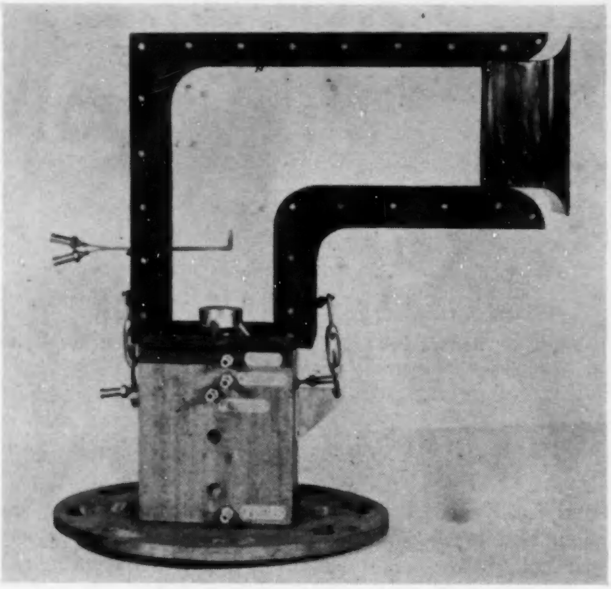
Originally the aircraft carburetor airscoop was merely a frontal air intake to help accelerate the air from flight speed up to its velocity through the carburetor. Now the airplane speed and slipstream speed have risen far above the velocities we dare use in the carburetor and manifolding, and they represent a source of energy that can have profound effect upon the whole intake function. It therefore seems worth while to begin by considering the whole intake system as one unit, of which the different sections



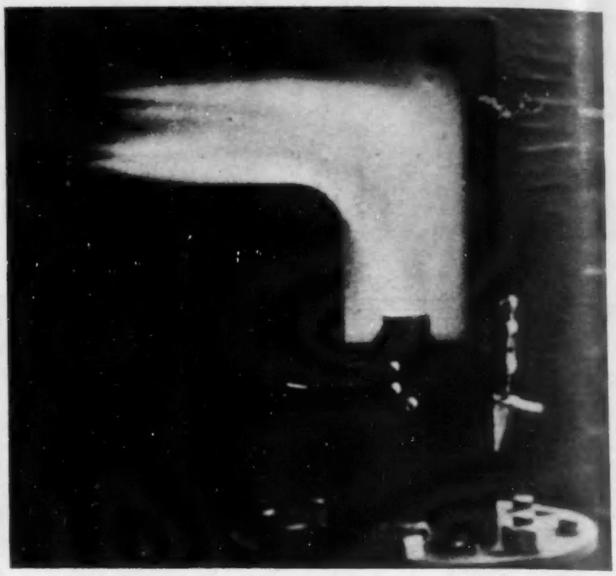
■ Fig. 2 – "Round" airscoop elbow constructed of wood spacers with sheet Lucite walls



■ Fig. 3 – Moving-picture "still" showing kerosene smoke flowing through elbow of Fig. 2



■ Fig. 4 – Transparent model of "sharp-corner" elbow



■ Fig. 5 – Smoke flow through elbow of Fig. 4

are mutually interdependent. In such treatment the main considerations should perhaps be:

(a) To obtain greatest possible ram and maximum *critical altitude* with least addition to the airplane head resistance (which is definitely in the domain of airplane research and design), and

(b) To obtain this objective with the least possible interference with the fuel and air metering functions (which brings together the airplane engineer and the carburetor engineer).

Fig. 1 is a sort of composite sketch of late practice in air intakes, and illustrates the varied and sometimes conflicting considerations of design. To indicate relative areas we have given at different points the air velocities of average practice, *at rated power at sea level.* At critical altitudes some of these velocities may be nearly doubled, and at light cruising powers they may be much decreased. Considerations of supercharger design make it desirable that the entrance diameter *A* be kept as small as possible. It is therefore necessary that the approach *B* to the impeller entrance be made free and direct; but, as this design tends to add appreciably to the overall length and weight of the engine, a generally accepted compromise is to change at *B* to a rectangular section of enlarged area, and to slant the passage at the angle *C*, less than a right angle, to decrease bend losses. We regard the transition shape from *B* to *A* as one of the most important and difficult elements of the intake system: first, to insure uniform and normal air flow into the impeller disc; second, to obtain similarly uniform delivery of fuel spray (when the fuel has been released anterior to this point). To avoid flow energy loss, it has now become accepted practice to maintain the generally rectangular proportions of *B* up through the carburetor, scoop elbow, and scoop duct.

Reverting now to the scoop entrance, this is preferably located where it will receive full air-speed ram, but far enough from the propeller that it does not feel the "spat" of air from each passing blade. Apparently the opening should be raised above the cowl somewhat to clear the boundary layer, and to maintain ram during climbing attitude. The area of opening should be such as to maintain during climb a ram value only a few inches below the indicated air speed impact pressure across the full section *F* of the carburetor entrance which, of course, requires a well-designed scoop elbow.

Because of the difficulty of knowing the distribution of flow rate when it is not uniform across a given section, and the similar uncertainty as to local air density, we have adopted the practice of using a *"nominal velocity,"* obtained by dividing the air volume flowing in a unit time, assumed at external air density, by the area of the passage section.

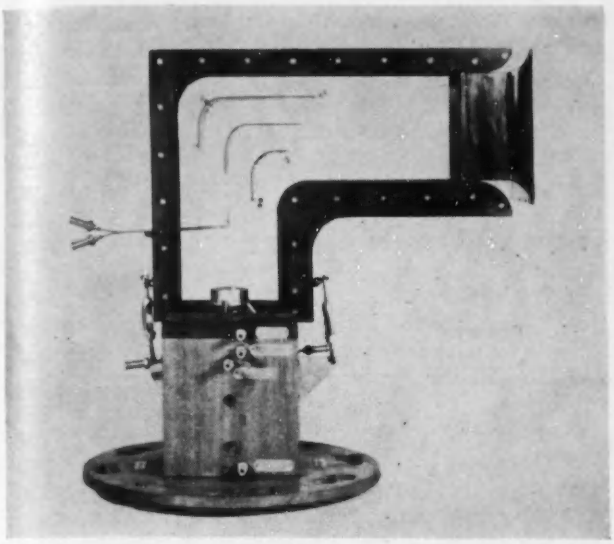
The considerations governing design of the scoop elbow will be discussed in later paragraphs. It will be noted that decreasing the angle *C* (Fig. 1) to favor the air entrance to the supercharger, has made the scoop elbow *G* angle more acute and less favorable to air flow.

As to the carburetor air passage, the important elements are the throttle, venturi throat, the air-speed metering elements therein, and the density compensator or aneroid. The way in which these elements are affected by *scoop phenomena in flight* is a major subject of this paper. As previously suggested, one of our main objectives is to achieve a practice whereby the metering of the carburetor as installed in the airplane in flight will be as nearly as possible the same as on the engine test stand.

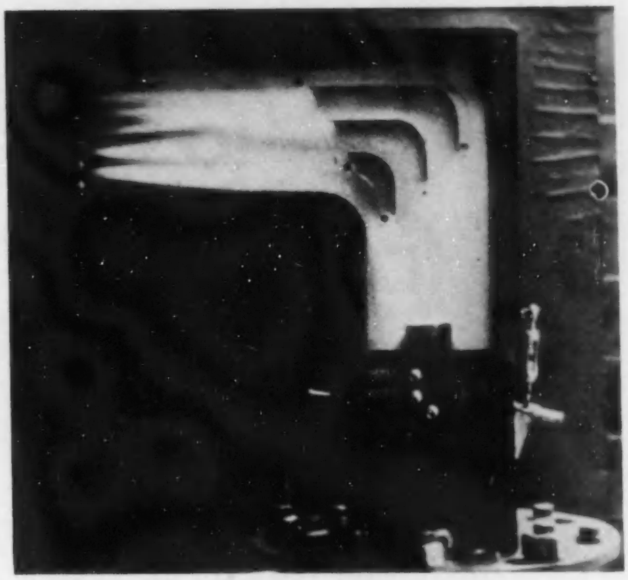
■ Elbow Effects

These effects have received major attention in past discussions of scoop design, and one fallacy in such treatment has lain in the assumption that what happened in the elbow on ground test, under *steady inflow* of air, was exactly what happened in flight. Actually, flight conditions sometimes seem to superimpose new effects upon those of steady flow. However, we will first discuss a few of the basic forms as to their steady-flow characteristics.

Fig. 2 shows a "round" elbow. The actual model was constructed of wood spacers with sheet Lucite walls. Fig. 3 is a still taken from a moving picture film of kerosene smoke flowing through this elbow (under conditions of



■ Fig. 6 – Scoop elbow of Fig. 4 with straightening vanes



■ Fig. 7 – Smoke flow through scoop of Fig. 6

suction-induced airflow), from which it will be seen that the stream followed the elements of curvature fairly closely. There is a slight tendency for the air to leave the inner radius. In airbox test this type of elbow disturbed the carburetor metering very little. Pressure measurements in flight on one airplane with this type of scoop showed a quite even pattern at the carburetor deck, and we had no evidence of appreciable departure from ground-test metering; this was with a short over-cowl scoop. On another airplane with this same general shape of elbow, but with an extension forward of some 3 or 4 ft, there was some metering disturbance, and not quite as even a pressure pattern on the carburetor top deck as with the previously mentioned airplane. We suspect that, if turbulence is excited in the scoop duct by constriction at the entrance or by oblique or turbulent flow induced by the propeller, the air stream can oscillate rather easily in passing around this type of bend; but we have nothing to prove this point.

In the test of the scoop models shown in the photographs, velocity pressure traverses were made at three levels above the metering elements of the carburetor. The velocities as observed are in good agreement with the flow pattern as indicated by the kerosene smoke. Velocity pressure was taken with a small pitot tube which is shown in the photographs of the transparent models in Figs. 2, 4, 7, 9, and so on.

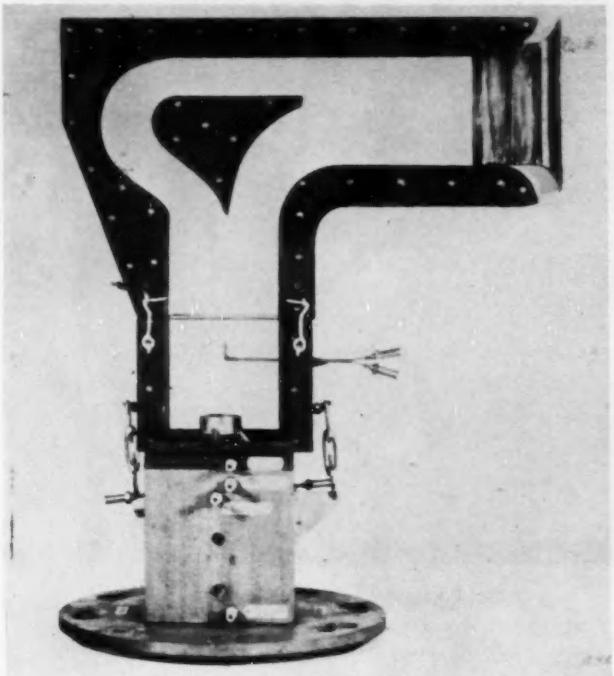
Fig. 4 shows a transparent model of a "sharp-corner" elbow, while Fig. 5 shows the smoke flow through it. It is difficult to appraise the velocity distribution when the stream lines are irregularly oblique, as they are in this case but the smoke figures indicate that the flow goes toward the back of the scoop just beyond the elbow. Presumably, if the scoop had had about two diameters more of straight run below the elbow, the flow would have nearly equalized across the section.

A scoop elbow similar to that shown to the left in Fig. 30 was tested in flight by one of our engine companies, and the pressure distribution observed across it, which is generally similar to that observed in the airbox. In this particular flight test there was a negligible disturbance of carburetor metering at the lower air velocities, but quite a bit at the higher velocities, as will be discussed later. We have other installations with this shape of scoop and metering disturbance has sometimes been reported.

Fig. 6 shows the previous scoop elbow of Fig. 4 with straightening vanes. Fig. 7 shows the smoke flow through this scoop. It will be noted that there is a pronounced high-velocity stream off the forward vane surfaces, which emphasizes the need for a vertical rise of preferably about six vane space widths between the lowermost vane and the carburetor metering elements. With this, there was negligible metering disturbance in the airbox. We have but one report of flight test with this shape, and in this no metering disturbance was reported. It will perhaps be of interest to compare Figs. 6 and 7 with Fig. 30 in later pages.

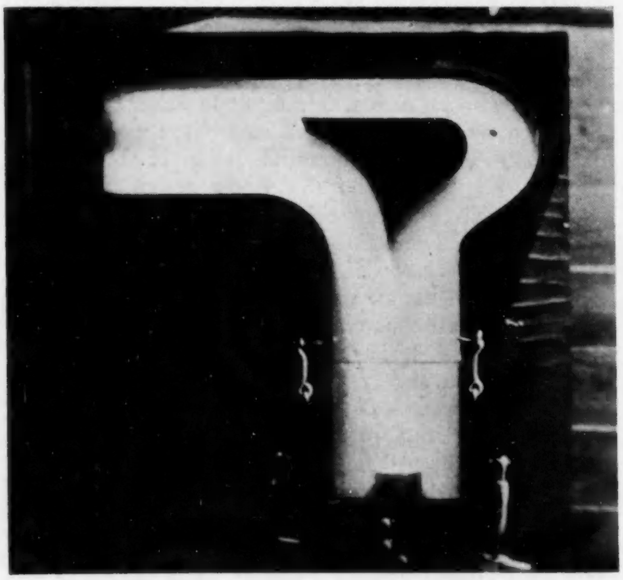
Figs. 8 and 9 show what we have called for obvious reasons an "island" scoop. The conception here was to create a central and symmetrical flow of air down to the carburetor by opposing the two halves of the entering air stream to each other as they enter the vertical down sweep. As the graphs show, there is a tendency for the flow to be strong down the center so that a considerable length of riser is necessary. Also it is difficult to balance the front and rear flows accurately against each other; so that airbox trials are necessary for successful design of this type of elbow. An application of this elbow design, shown in Fig. 36, has given quite satisfactory performance in flight, with no reported metering disturbance. We are inclined to believe that a scoop of this sort, that compels division and remixing of the air stream at the scoop elbow, is less affected by propeller-induced turbulence than are the single-passage types; but again we have no definite proof of this.

Figs. 10, 11, 12, and 13 show characteristics of the "acute-angle" scoop elbow such as is illustrated in Fig. 1. As previously mentioned, turning vanes tend to give a high-velocity stream immediately below the vanes. In order to break up this high-velocity stream, the vanes as shown in Figs. 10 and 12 have been made discontinuous. Well around the corner, the vane has been broken, and the lower part moved slightly forward. This produces an additional number of high-velocity streams of lesser mag-



▪ Fig. 8 – Model of "island" scoop

nitude which will recover and give uniform distribution in a shorter length. It also will be noted that a slight hump has been installed at the back of the scoop just below the turn. This hump or chute breaks up the high-velocity stream which follows the outer radius. These pictures illustrate the point of carrying the entrance end of the vanes forward across the warm air opening so as to equalize the pressure at the carburetor deck; as is well known, most warm-air valve constructions are sadly deficient in this respect. We have no reports of flight tests with this shape of scoop with vanes in it; however, we did find in airbox test that use of the vanes regained at the carburetor top deck 1 in. hg of pressure that was lost by the acute



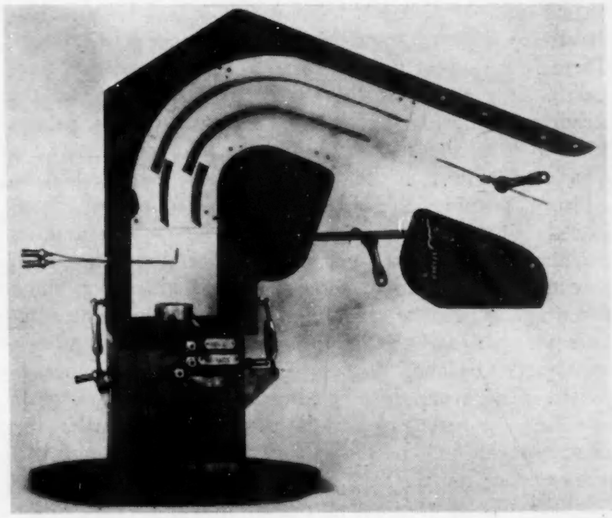
▪ Fig. 9 – Smoke flow through "island" scoop of Fig. 8

angle without the vanes. Since this pressure loss would be magnified by the supercharger, there is reason to believe that, other things being normal, use of the vanes should give at least 1000 ft gain in critical altitude and possibly considerably more, with an acute-angle elbow like this.

How Uneven Air Flow Disturbs Air Metering – We have seen that the different elbow shapes result in uneven velocity distribution across the face of the air stream, and we have also found that these irregularities continue down into the carburetor. The manner in which they affect metering is of some interest.

In essence, present-day carburetors meter by *measuring the velocity* of flow *in an air stream of known area,* with correction for the density of the source of flow. The carburetor structure thus involves a constriction, or venturi tube, to determine the area of the air stream, and one or more elements which respond to the air-stream velocity and transmit to a fuel-metering mechanism some measure of the pressure differential between the air-stream *impact,* or *total pressure,* on the one hand, and its *velocity* depression, on the other.

Through a large range of air flow such a differential



▪ Fig. 10 – Acute-angle scoop elbow with discontinuous vanes – full-cold heater-valve position

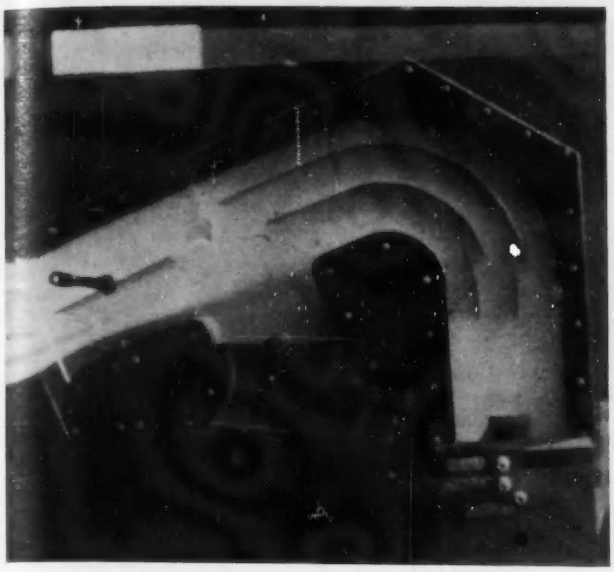
varies with the air velocity squared, times the initial air density. The fuel flow follows a similar relation to its metering head, so that, if air and fuel differentials are either equal or proportional through the forementioned range, the following will be true:

(1) $(\text{Air Velocity})^2 \times (\text{Air Density})$ varies as $(\text{Fuel Velocity})^2 \times (\text{Fuel Density})$

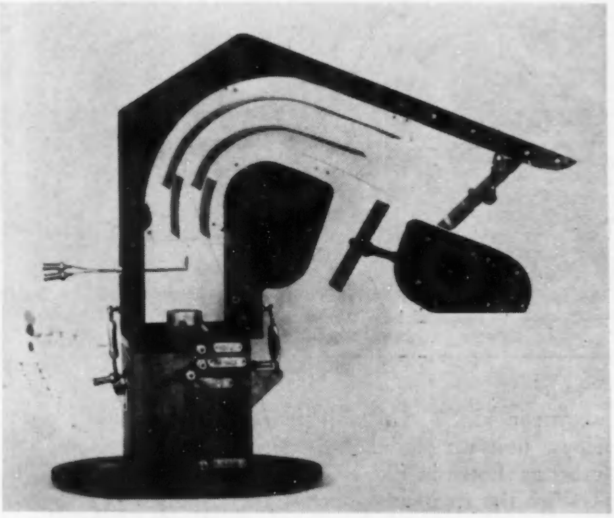
(2) $(\text{Weight Flowing}) = (\text{Velocity}) \times (\text{Area of Stream}) \times (\text{Density})$

$$(3)\quad \frac{\text{Fuel Weight Flowing}}{\text{Air Weight Flowing}} = K \times \frac{\text{Fuel Stream Area}}{\text{Air Stream Area}} \times \sqrt{\frac{\text{Fuel Density}}{\text{Air Density}}}$$

That is, at constant densities, a constant fuel-air ratio



■ Fig. 11 – Smoke flow through acute-angle scoop of Fig. 10



■ Fig. 12 – Acute-angle scoop elbow with discontinuous valves – full-hot heater-valve position

may be obtained from fixed-size air and fuel orifices, and any desired regulation of the fuel-air ratio may be obtained by corresponding change of the fuel orifice size. Also, the correction for varying air density is as the square root of such density change, and, under any density, remains constant for different air velocities (until the critical range is reached, as will be discussed later).

Fig. 14 shows a number of different constructions for obtaining air-velocity differential. It is not generally appreciated that the *impact* or total pressure reading remains almost constant at different positions along the venturi tube from entrance to outlet, and is equal to the *static* pressure at an infinitely large entrance area; while the *downstream* pressure is but little different from the *static* at any given point along the stream. Thus, in Fig. 14, *B* gives but little greater metering differential than *A*. The "boost-venturi" construction *D* gives a higher metering differential for the same loss than do the others, and also possesses important advantages in the critical range, as will be discussed in detail later.

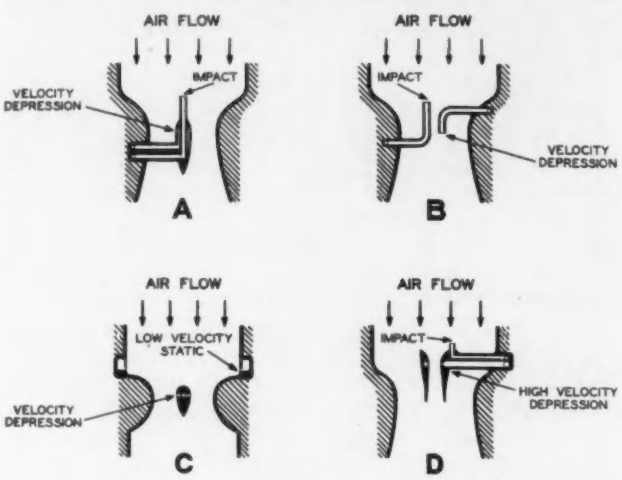
When we come to examine the effect of the mildly uneven air flows of the elbows upon the elements of Fig. 14 (mild as compared with some disturbances to be described later), we see that, if either the pressure or depression elements, or both together, are located other than in a point of ***average of mean square air velocity,*** the metering differential will vary from that generated by the same air weight flowing in a uniform stream. Actually, we do not find much difference among all the forms shown in Fig. 15 so far as scoop sensitivity is concerned; some are slightly better with one elbow, some with another. However, this steady-flow bend effect is not a serious drawback in our effort to give the airplane best performance, because we can reproduce it on the ground, on the engine test stand, or in carburetor airbox test, and modify either elbow shape or carburetor setting as needed. Of the two, it is preferable to modify the elbow so as to keep the carburetor setting uniform with that of the same engine on another airplane.

It should be noted that, in ground test without a ram duct, an elbow should have a funnel placed upon its entrance to reduce entrance constriction and give the same flow lines as would exist with ideal air-speed ram.

It also has been found that the sensitivity to elbow shape can be reduced by multiplying the number of air-speed elements. Obviously there is a limit to the number of these that can be employed without obstructing the air flow; also, mere multiplication does not correct completely, because of the velocity square relation. Fig. 15 shows a variety of scoop elbows which give negligible variation on the metering with the two ranks of air-speed elements of the type shown in *D* of Fig. 14.

Fig. 16 shows two sources of disturbance that should be avoided. The left-hand view shows the air stream constricted but still including the air-speed members, which makes the carburetor meter-rich. The right-hand view shows an obstruction, such as an alcohol tube, creating a

■ Fig. 13 – Smoke flow through acute-angle scoop of Fig. 12



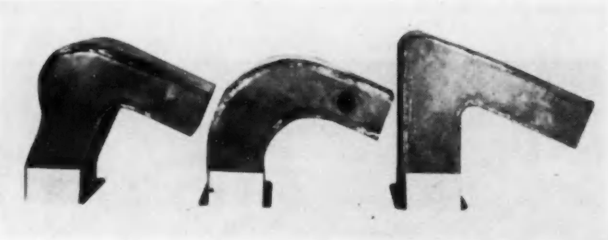
■ **Fig. 14 – Different arrangements of carburetor elements for measuring velocity differential through venturi-tube throat**

"lee," or dead area over the velocity metering member, which causes wide changes in mixture.

Disturbance from Propeller Slipstream

We come now to one of a series of phenomena which have been bothersome in carburetor work because the action in flight did not reproduce in ground test, namely, the propeller slipstream effect. After much work we found that, under certain conditions, the propeller slipstream did not expand to lower velocities in the scoop duct as shown in Fig. 1 but, instead, continued at its original high velocity as a turbulent and discontinuous flow, resulting in an irregularly shaped air stream to and through the carburetor which disturbed the metering considerably.

This particular trouble appears to be confined to scoops having their inlet close to the propeller. It is more marked

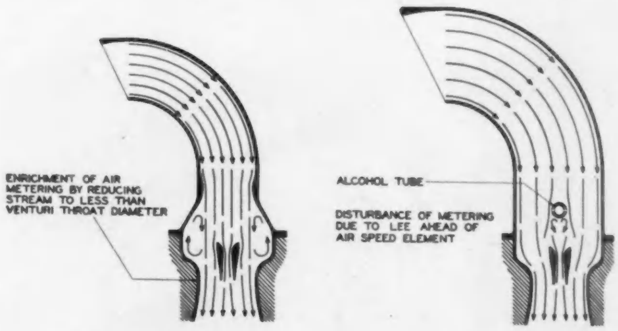


■ **Fig. 15 – Variety of scoop elbow shapes which do not affect carburetor metering when multiple air-speed elements are used. This result was obtained under airbox steady flow and may be different in flight**

when the slipstream velocity is high with reference to the airplane air speed – as, for instance, when "reving up" on the ground and during take-off. If it occurs at all in steady level flight, it will probably be in high blower, and with high propeller pitch. Thus far we have found no means of overcoming this effect with anything we can do to the carburetor alone. Apparently we must either tolerate it, or remove the scoop entrance from the direct attack of the propeller-blade impulses.

When we first began to surmise that the slipstream was affecting the carburetor metering, we made the rather simple check illustrated in Fig. 17. On a Cyclone engine mounted on a rigid stand with a fixed-pitch club and no cowling, we mounted a round-corner airscoop, turned forward toward the propeller, with extensions of different lengths. With this set-up we tried the three different types of carburetors which have been used on this engine in airline service. The three charts show the pounds of fuel per hour versus engine rpm with the different scoop lengths, each carburetor's mixture adjustment being kept constant throughout its test. It will be noted that, *with this same elbow bend,* change of distance from the scoop entrance to the propeller blades made a pronounced difference in the fuel flow rate with all three carburetors.

We next tried to analyze the air flow through the scoop to find what there was about it that so greatly affected the carburetor metering. For this we used the set-up shown in Figs. 18 and 19, which show respectively the shortest and longest scoop extensions, out of several lengths which were tried on three different elbows: the right, or sharp, angle one shown; a round-bend scoop; and another form

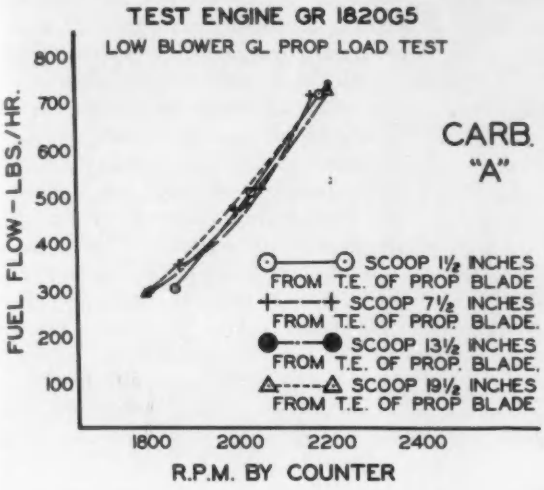


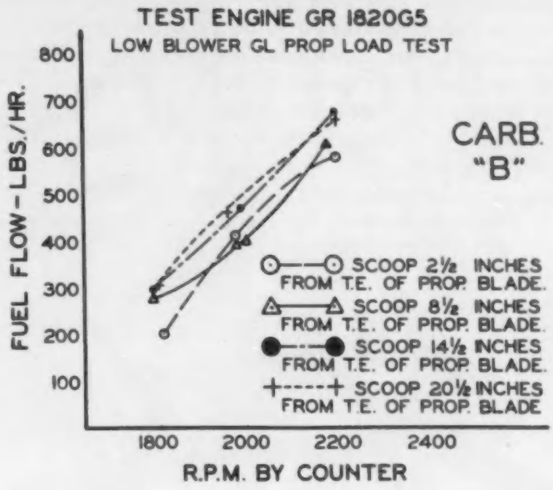
■ **Fig. 16 – Details of scoop construction near carburetor which disturb air-velocity metering**

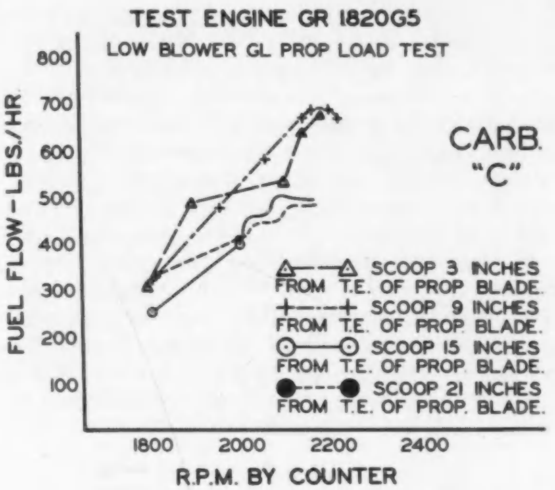
shown in Fig. 8. On both the entrance and outlet of the elbow unit we placed grids of impact, or total pressure, tubes as shown in Fig. 20.

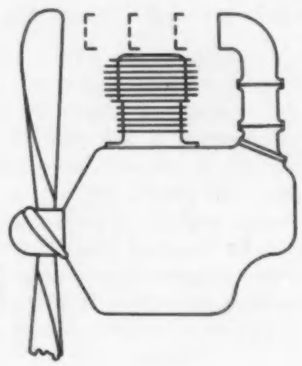
With the sharp-angle elbow, the impact pressure distribution with different scoop extensions is shown in Fig. 21. With the short stub entrance, 19½ in. from the rear propeller edge, the impact pressure distribution was relatively uniform, not varying more than from 2 in. + to 1 in. – of H_2O at different points on both grids. With the next longer extension, 13½ in. from the rear propeller edge, the variation in pressure distribution was more marked, amounting to 6 in. H_2O. With the next longer extension, 7½ in. from the propeller, the variation was more sharply localized and reached 8 in. H_2O. With the scoop extended to 1½ in. from the propeller, the pressure was a little higher, but the distribution about the same. Further study and observation with tufts seemed to indicate that the slipstream just off the propeller blades had a pronounced spiral direction which piled up pressure in the far side of the entrance of the longer scoops, and that there was also developed a spiral curl down the two opposite corners, which condition sometimes was maintained symmetrically at the scoop base flange and sometimes was reversed.

Fig. 22 shows the pressure distribution obtained with the three different scoop elbows with extensions 7½ in. back of the propeller edge. It will be noted that, with all these



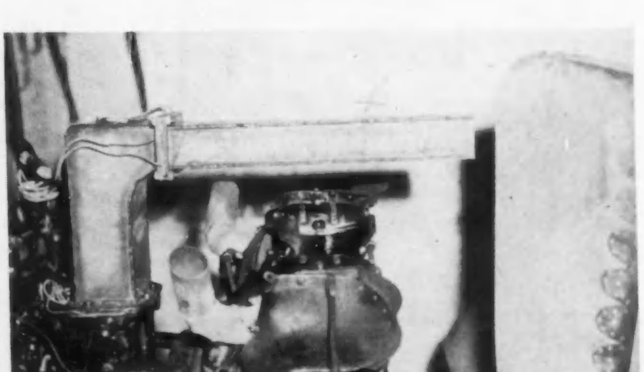






■ Fig. 17 – Disturbance of metering due to propeller slipstream – three different types of carburetors with same scoop elbow – scoop entrance placed at different distances from fixed-pitch propeller trailing edge

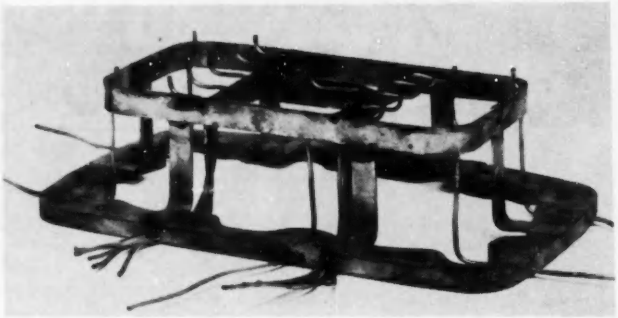
■ Fig. 18 – Sharp-corner scoop elbow with short extension, as shown in Fig. 21



■ Fig. 19 – Sharp-corner scoop elbow with long extension, as shown in Fig. 21

shapes, there was very little difference in pressure pattern.

Pressure Pulsations Caused by the Propeller – In addition to the turbulence and disturbance of air stream pattern just noted, we have had repeated evidence that the propeller blades created a pressure-time pulsation in the scoop duct, most marked when the scoop opening was close behind the propeller rear edge. With one airplane, a definitely audible, very low-pitch note was reported by the pilots, which disappeared when the scoop was shortened some 10 in. In the effort to find out more about this phenomenon, we have been developing a pressure indicator especially adapted to a study of these delicate and rapid pulsations. The equipment is illustrated in Fig. 23 and its scale size may be judged from the accompanying batteries shown. The special equipment consists of a power unit, an amplifier unit, special pressure pickup gages, an oscillograph, a recording camera, and remote control for operation from the pilot's cockpit. It is constructed to operate either in the laboratory on 115-v, 60-cycle alternating current, or in an airplane on 12-v direct current. The pressure pickup element was developed for us by the General Electric Co. laboratories at Schenectady and has proved remarkably stable and consistent. As would be expected with so delicate an instrument, a major problem has been that of interpreting all the disturbances recorded.



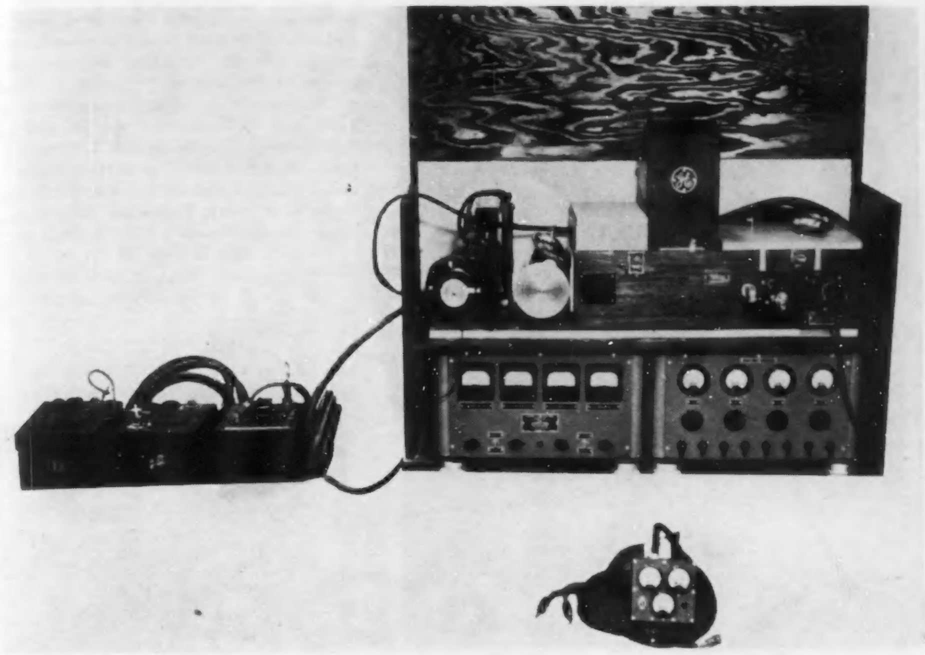
■ Fig. 20 – Impact grid used in tests of Figs. 21 and 22

So far our work with this device has been confined to checking and calibrating in our laboratory and on an engine on a ground-test stand. Fig. 24 is a typical record as suggesting the characteristic and scale of the indications. The lowest line was obtained from a contact breaker driven off the gun synchronizer on the engine, and timed to show a break each time a propeller blade passed the front of the airscoop. The second line is a timing wave fed directly from the power unit with 1/100 sec frequency. The scoop pressure pulsations are clearly marked and, in this test, quite regular. It will be noted that they time exactly with the propeller blade passage. Their total amplitude is about 14 in. H_2O pressure. Apparently, in this case, the propeller was the chief source of pressure oscillation.

Fig. 25 shows another record at 1800 rpm ground propeller load (the previous one was 1200 rpm). In this record the pressure phenomena are more complex and there is evidence that some other frequency, possibly the engine suction strokes or the natural frequency of the scoop as an organ pipe, was striving for ascendancy with the propeller beats. However, the propeller impulses were clearly dominant. It is too early to say just how much information we may hope to gain from this sort of test. In the past we have gone through a few experiences where we are quite sure it would have been valuable, particularly some of those to be described in the following section.

As regards this propeller-blade effect on our test stand, we did a great deal of work with the carburetors, changing their metering elements, and so on, in the effort to find a cure or offset to this high turbulence in affecting the carburetor metering, and nothing we tried reduced the maximum disturbance to less than 10% change in air-fuel metering between the long and short extensions. We have

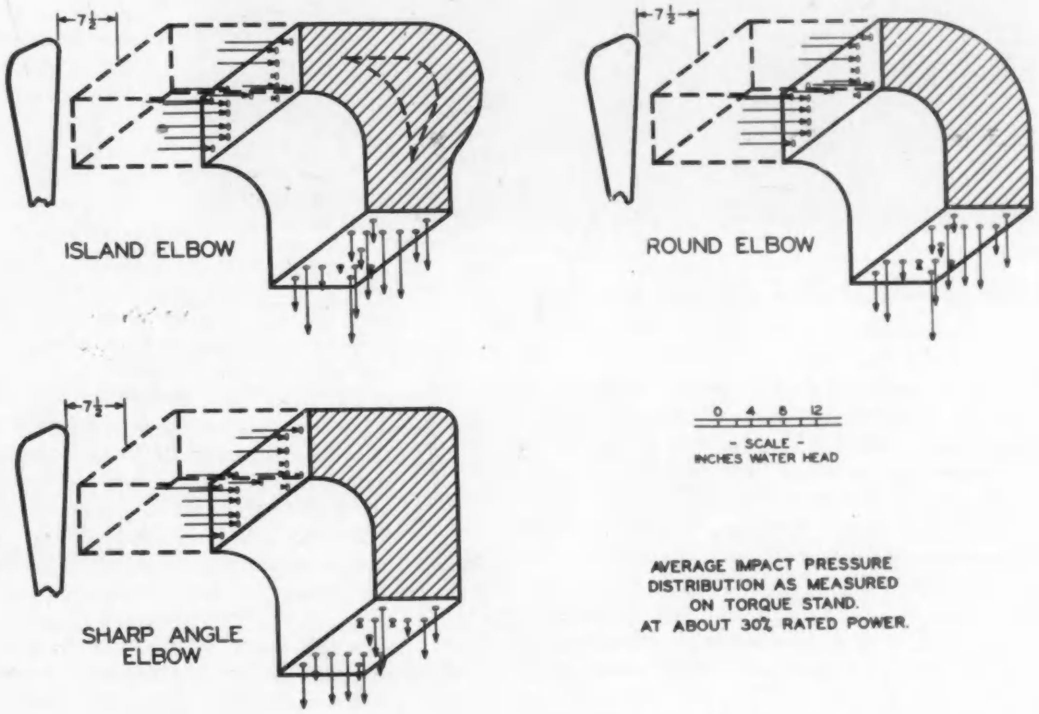
■ Fig. 21 – Increase of irregularity of pressure distribution at airscoop and carburetor entrance as the scoop opening is brought toward the propeller



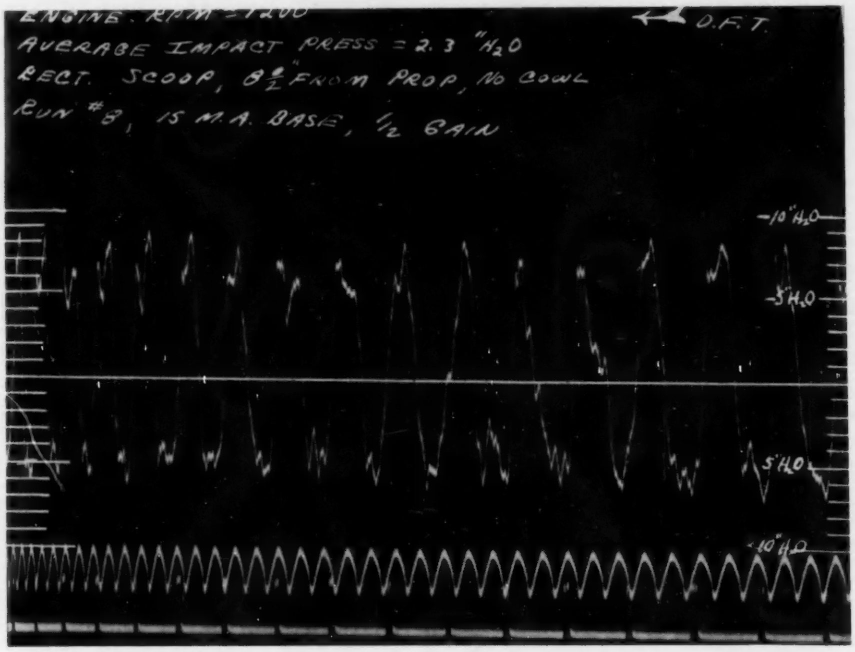
■ Fig. 23 – Oscillograph apparatus for indicating and recording pressure pulsations in airscoop, carburetor, and intake manifold – See records on Figs. 24 and 25

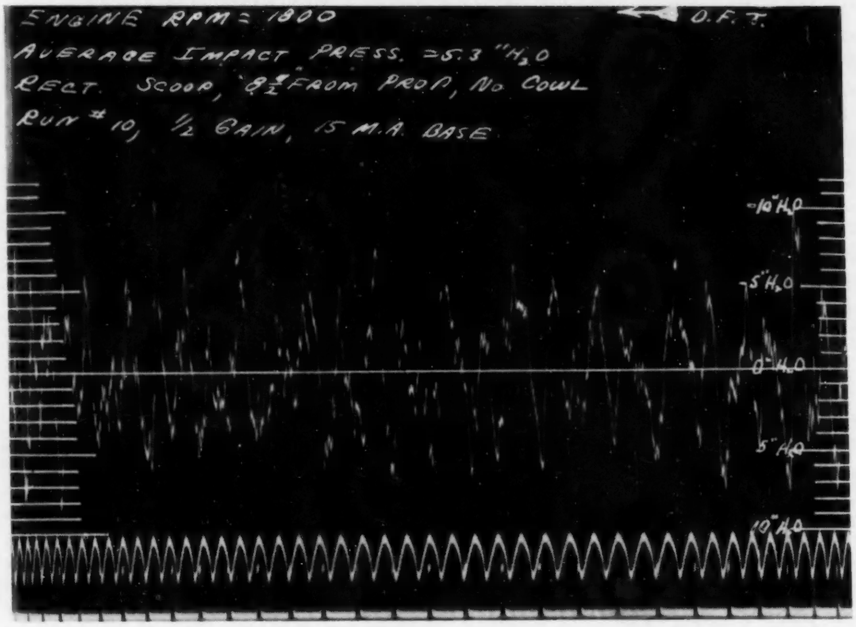
therefore more or less given up hope of ever taking care of this sort of disturbance by anything we can do to the carburetor. We believe, however, that this effect is confined to take-off and "reving up" since we did not find a similar pressure variation in any of our flight tests, some of which were made with the identical grid units used in these experiments.

If there is good reason to bring the scoop entrance up close behind the propeller, we would suggest that the opening be flat radially from the propeller center and long in its circumferential direction, as shown in *A* and *B* of Fig. 26, so that the propeller-blade width does not close off or trap a pressure impulse in the scoop as it goes by. In the average of our experience we have had less change of carburetor metering between flight and ground test with these than with Type *C*. To date we have not been able fully to identify any given metering disturbance in flight as solely due to a scoop opening close to the propeller. But we are rather afraid of these installations and we hope that future aerodynamic research will develop adequate ram characteristics without the necessity of bringing the carburetor scoop opening so close.



■ Fig. 22 – Pressure distribution at carburetor entrance with different scoop elbow shapes – scoop entrance near propeller





■ Fig. 24 (left) and Fig. 25 (below) – Records of impact pressure pulsations in scoop with air entrance 8½ in. from propeller trailing edge using a fixed-pitch club on a rigid torque stand at 1200 and 1800 rpm in Figs. 24 and 25, respectively. Breaks in the bottom lines indicate times when propeller blade is passing scoop entrance. The middle line is the time track, frequency 100 per sec. Note the amplitude of 15 in. H_2O pressure in Fig. 24

■ Enrichment at High Altitude and Power

The "scoop effects" thus far noted have been relatively insignificant. We now come to a disturbance which has been more annoying because of its consistent refusal to appear at all in any sort of ground test, although occurring fairly often in flight. This has made it particularly difficult to deal with under our present habits of procurement, which prescribe elaborate details of ground test with little provision for check or development work in flight. This disturbance manifests itself as an increase of air-speed metering differential at high altitudes, high powers, and high speeds, and is superposed upon the normal enrichment due to decreasing air density. It appears to have some of the characteristics of *critical velocity flow* and to be, in some measure, due to the elbow and propeller disturbances previously enumerated. Because it occurs only in flight, and because modern airplanes have not been available for a systematic research program, our knowledge on this subject is at present fragmentary.

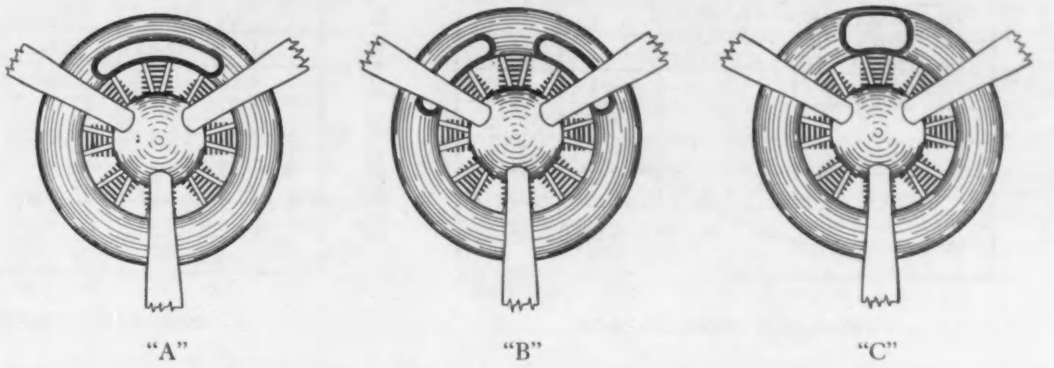
Referring first to basic air-speed metering characteristics *under steady flow,* well-known and repeatedly checked in air box test, Fig. 27 shows on the upper left-hand chart the velocity metering depression and also the loss or flow resistance obtained with different air flows at sea level with both a single venturi and boost venturi of standard proportions. In each metering curve its lower part is a true parabola. (The continuation of this true parabola has been marked as a dotted line.)

In the range of the true parabola, the fuel-air ratio tends to have a constant characteristic, as previously explained, but, as the metering depression rises above the velocity square ratio, the fuel-air ratio increases, and we call this the *velocity enrichment.*

For this experiment the two venturi were selected to have about the same "loss" at sea level. The "single" has a lower metering suction than the "double," or boost form, which objection could have been helped by making its throat smaller, and carrying its expansion cone to the same diameter as before. But this change would have given a higher throat velocity and increased the *velocity enrichment* and loss, which are already higher than that of the boost venturi.

The right upper chart shows similar values at 20,000 ft – *under steady flow.* It will be noted *first* that the depressions have increased inversely as the air density for a given weight air flow; *next,* with each venturi form, the departure from a parabola is greater at a given weight air flow than at sea level; and, *finally,* the *critical point* of the single venturi, where the metering force and loss rise rapidly, is clearly shown to occur at about 7800 lb of air per hr.

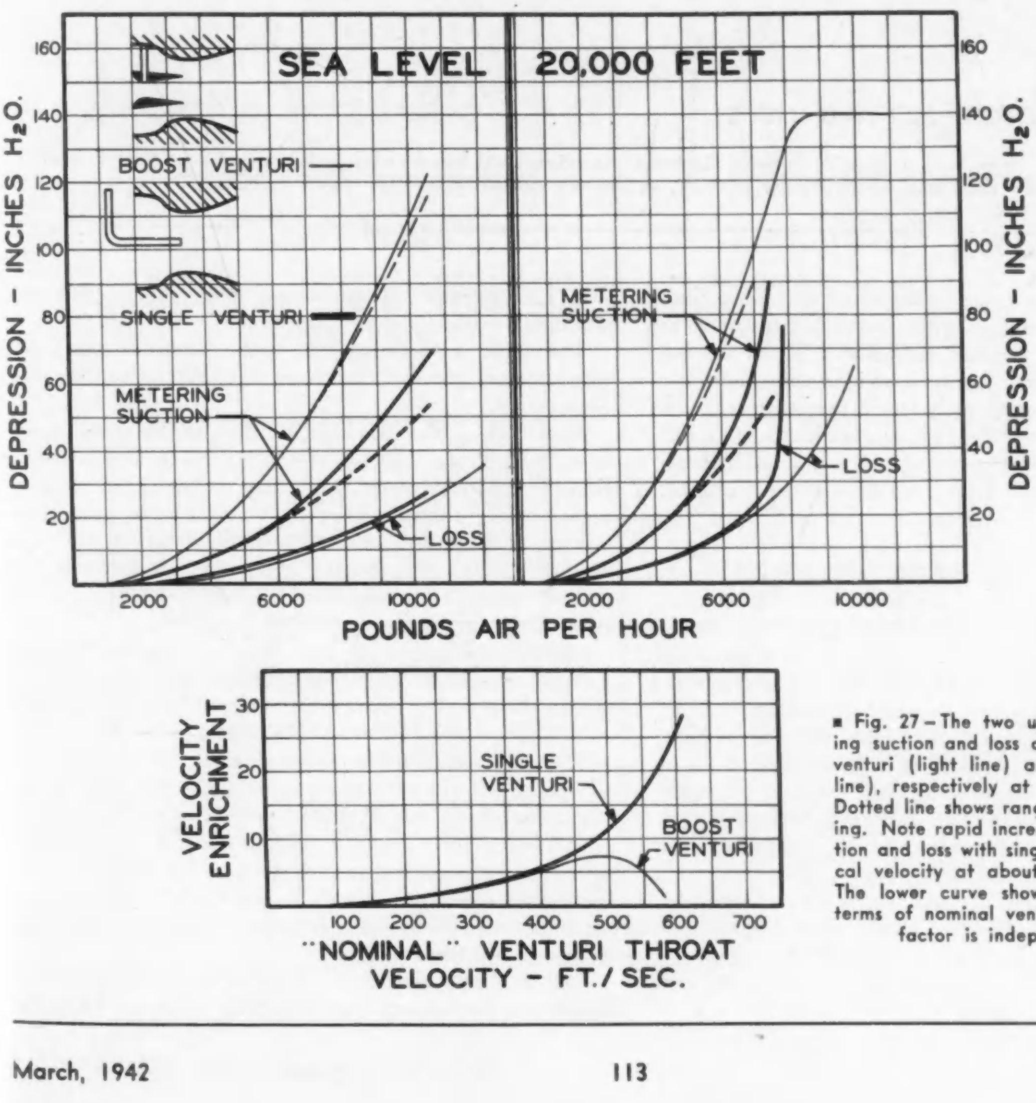
If, instead of plotting the air flow of these two curves in weight units, we use velocity, and correct for the density, we find that the different altitudes coincide, and we obtain the characteristic *velocity enrichment* relationship of the lower chart, which brings to light a little-known



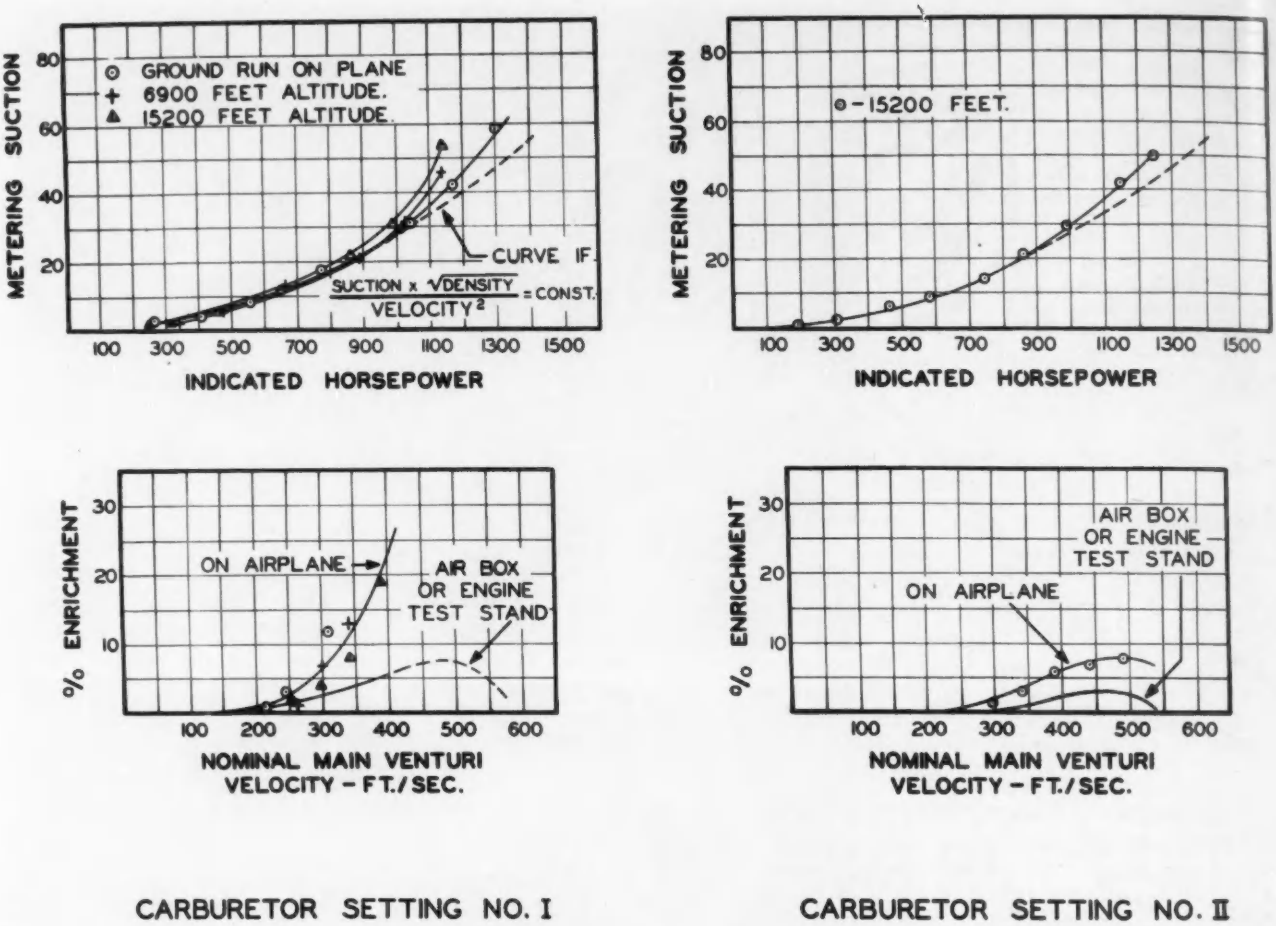
■ Fig. 26 – Front cowl air intakes with varying degrees of propeller-blade interference

characteristic of the boost venturi construction. The boost, or smaller member, is really a channel in parallel with the conical entrance of the larger member and, as we see in the case of the single venturi in the upper right-hand chart, any increase in enrichment is accompanied by an increase in "loss," or flow resistance. The result is that an inherent compensation takes place, under steady flow at least, and the depression in the small or boost venturi bears a constant relation to that in the larger one until the critical condition is reached in the boost venturi throat, after which its depression stays constant under increasing air flow so that the fuel-air ratio starts down. The point of starting down depends somewhat upon the proportions of the boost venturi; in this case it comes at about 500 fps *nominal velocity* in the main venturi – that is, the velocity which would exist in the main venturi throat if the air were at entrance density.

The curve here given has been repeatedly checked by



■ Fig. 27 – The two upper charts show metering suction and loss characteristics with boost venturi (light line) and single venturi (heavy line), respectively at sea level and 20,000 ft. Dotted line shows range of true velocity metering. Note rapid increase of both metering suction and loss with single venturi, reaching critical velocity at about 7800 lb of air per hr. The lower curve shows percent enrichment in terms of nominal venturi throat velocity. This factor is independent of altitude



■ Fig. 28 – Difference in carburetor metering between flight and ground test, with two different carburetor settings. Note in each case the increased enrichment in flight, apparently increasing with air intake velocity

different authorities and may be regarded as a basic characteristic. It should move slightly to the left inversely as the square root of the absolute temperature if the abscissa is *velocity;* but, if the base line is air-weight units, the curve should move to the right with lower temperatures. All this, of course, is under steady-flow conditions. On this basis, we have selected for our carburetor practice a maximum air velocity of 400 fps for boost venturi constructions, and 300 fps limit when single venturi are used.

Velocity Enrichment in Flight – The foregoing is preliminary to consideration of *velocity enrichment* in flight. Fig. 28 in the upper left shows curves of metering suction, each at a constant altitude, plotted against *indicated horsepower.* By using a known factor of reasonable accuracy we can compute the air flow itself; and, if we assume this factor constant, we can apply the parabola characteristic to these curves as was done with Fig. 27. Then, if we plot the departure from the parabola as ordinate against nominal main venturi air velocity as abscissa, we have a picture of the departure from normal air metering. Note that this method eliminates all other carburetor characteristics and variables from the picture. It does require that:

a. The temperature and pressure of the air be accurately measured:

b. The fuel-air ratio should be within the range of maximum power; otherwise, our air consumption per horsepower factor will change.

c. The altitude compensation must remain constant throughout each fixed altitude run.

The scale of accuracy, of course, depends upon the torquemeter, and the correction from brake to indicated horsepower.

Nevertheless, the conclusions indicated by these curves have been borne out by exhaust-analyzer and fuel-consumption measurements, and may be regarded as clear indications of a trend. They show rather clearly that there is a tendency toward enrichment with *increasing air intake rate,* of which the functional significance might be in the venturi system, the scoop duct, or (rather unlikely) in the supercharger entrance.

Though the curves do not show it, flight-test reports sometimes indicate greater enrichment in climb attitude (when the propeller disturbance referred to in the previous section might possibly have been more active).

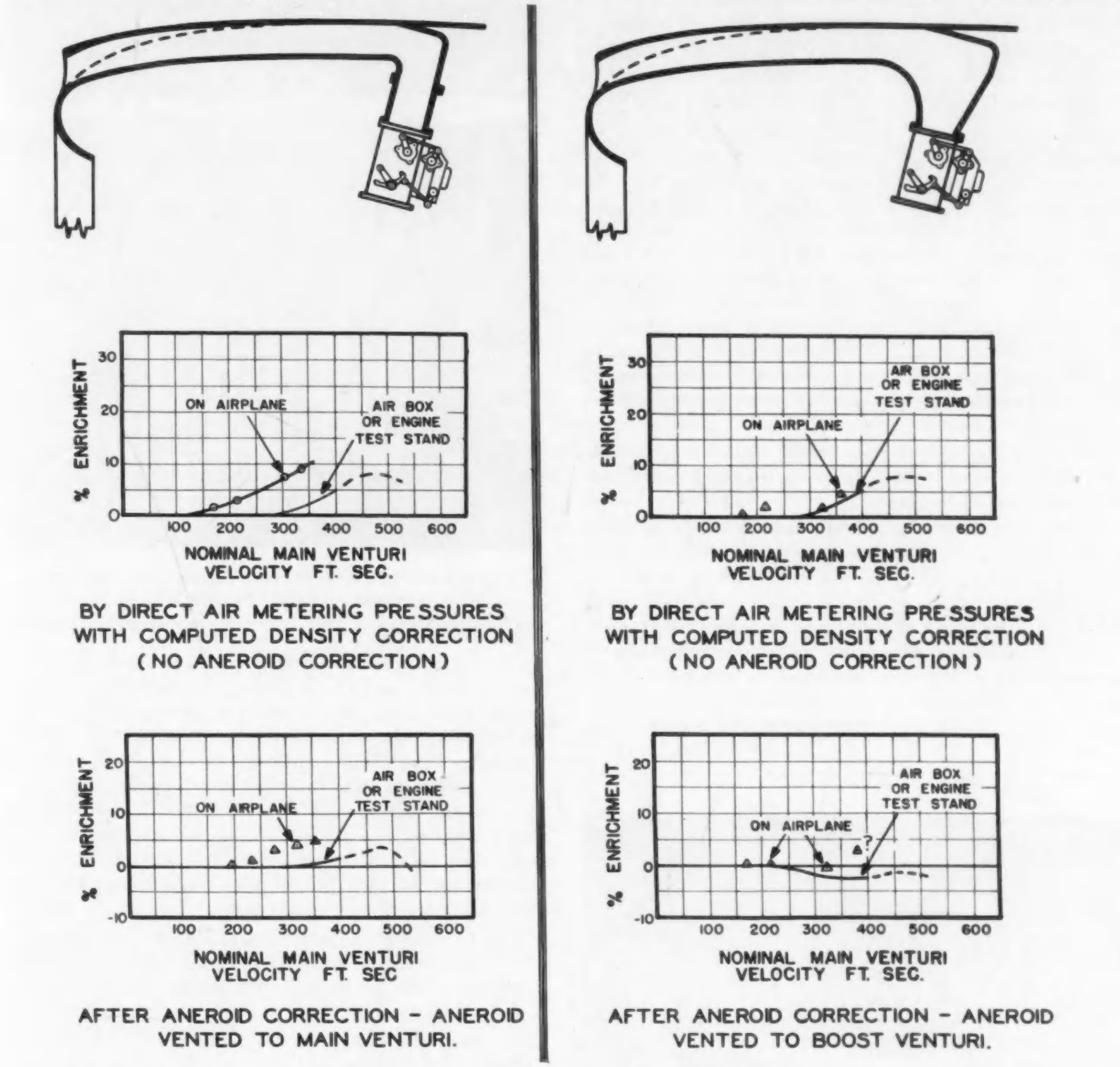
The lower left-hand curve shows the enrichment versus air intake velocity, as reported in flight; also that of the carburetor alone under steady flow in ground test. The right-hand charts show a similar pair of indications after we made a change in the carburetor to throw it leaner at the higher velocities. It will be noted that the change in steady-flow characteristics is reflected and, if anything, augmented in the airplane.

Fig. 29 shows results of another type of flight test in which the horsepower was left fairly constant while the

ship climbed. This test involves the same scale of uncertainty as the one previously described and, in addition, rests upon the promptness and accuracy of the aneroid in passing from one altitude to the next. Yet these readings were also borne out by those of exhaust analyzer and flow meter.

They also show the general characteristic of velocity enrichment to a much greater degree than does the carburetor alone. Again a minor change in the steady-flow ground metering of the carburetor shows a major change in the characteristics in flight. Incidentally, the pressure pattern in the airscoop, though taken at only a few points, is better and more even than that which we obtained in suction tests with the scoop shown in Fig. 4. This seems to be generally true with all the scoops of which we have taken patterns in flight.

Perhaps the most serious element of the difficulty shown in the foregoing tests is the fact that the larger scale of disturbance occurs only on the airplane. Thus far we have been unable to reproduce it in the laboratory. We have put the ram from a blower on the entrance of the airscoop and graduated it to give the same pressure pattern on the face of the carburetor as was measured in flight; yet we did not get the enrichment. We have tried in various ways to



■ Fig. 29 – Velocity enrichment in flight with two different airscoops. Upper enrichment curves are derived from direct observed *velocity depressions*, by variation from:

$$\frac{\text{Suction}}{(\text{Nominal Air Velocity})^2 \times \text{Air Density}} = \text{Constant.}$$

Lower enrichment curves are derived from observed *metering suction*, after aneroid correction, by variation from:

$$\frac{\sqrt{\text{Metering Suction}}}{\text{Lb Air per Hr}} = \text{Constant}$$

augment the entrance effect at the scoop opening without appreciable effect. It was our hope that we might record in flight the scoop duct pressure pulsations with the indicating device previously described, and then reproduce these in a laboratory, but we have not yet achieved this result.

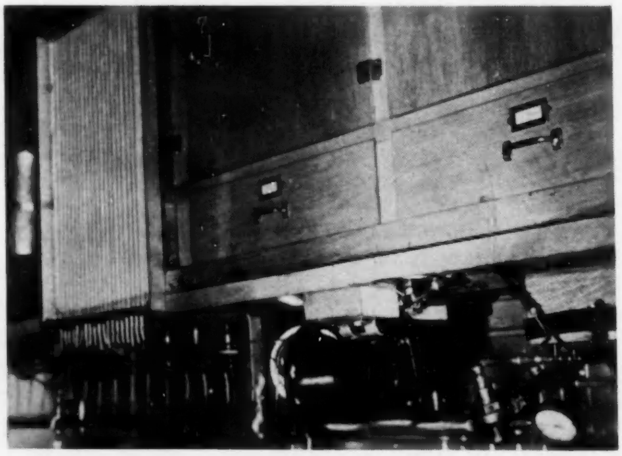
From general experience it seems that the disturbance is associated somewhat with displacement of the air stream reaching the carburetor. It seems to be also associated somewhat with propeller disturbance. I do not recall that we have had any disturbance of this sort in turbo installations. As a matter of fact, in the present extent of our knowledge, it seems that there could only be two reasons for this change of air metering characteristic:

(a) Periodic variation of flow rate: since the depression varies as the momentary velocity squared, the integrated depression will be greater in pulsating than in steady flow for the same average mass flow per unit time.

(b) Diminution of the air stream to fill only part of the main venturi throat. This condition could result if the air stream should continue through the scoop duct and carburetor at the initial slipstream velocity or plane air-speed velocity, provided this latter is higher than the normal venturi throat air speed required for that horsepower and altitude. This has been the case in nearly all the circumstances under which enrichment was reported. We have not had an opportunity to make flight tests comparing single and boost venturi, but by reference to Fig. 27 it will be noted that the enrichment has the characteristic of the main venturi depressions, rather than those of the boost venturi.

Whatever the cause of this peculiar disturbance, its cure will probably prove simple, once we learn the characteristics of the actions involved.

It is believed that to develop the nature and basic causes of this phenomenon will require some weeks of test in an airplane especially set aside for this purpose, using the methods and equipment already referred to. The work should be done under the direction of carburetor research engineers and should be accompanied by ground check on various points as the need develops. It is suggested that this is a very suitable and proper occasion for aid from the military services or the government research organizations.

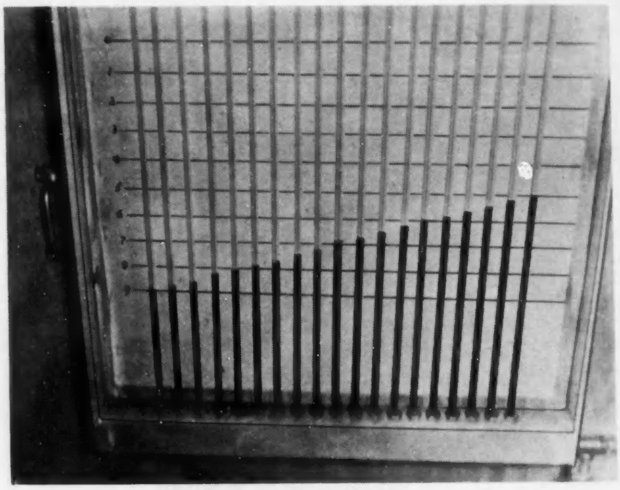


■ Fig. 30 – Demonstration of resonance waves with intermittent air blast directed squarely against long air tube

■ Pressure Wave Effects

Nearly every airplane builder has evidence at one time or another of the existence of pressure waves in the intake air system. We have already referred to periodic disturbances excited by the propeller, but we have had others occurring only at high air speed and independent of engine speed. We have had others arising from supercharger surge and, in one case, from turbo supercharger surge.

We already have mentioned the case where the pulsations in the scoop gave a definite low-pitch note which disappeared when the scoop was shortened. Another interesting instance occurred when a spacer between the scoop elbow and carburetor blew out, and the engine ran much smoother afterwards at altitude. I believe that thereafter a $2\frac{1}{2}$-in. hole was put regularly in the scoop at the elbow to "untune" the scoop pulsations. In one recent case, a



■ Fig. 31 – Distribution of pressures along tube in Fig. 30 with uninterrupted blast into tube

metering disturbance occurred at nearly closed throttle in the carburetor, and the frequency was definitely that of an organ pipe, closed at one end and having a length equal to that from the scoop entrance to the throttle valves of the carburetor. This frequency, about 50 to 55 cycles per sec, happened to coincide with the natural response rate of our aneroid which had been previously damped to withstand mechanical engine vibration of very much higher rate. We prepared to install the oscillograph device previously described but, meanwhile, it was fortunately found that this trouble could be mitigated:

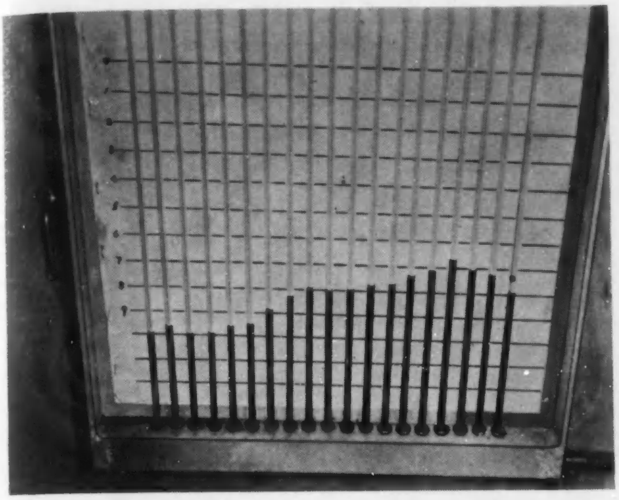
(a) By changing the scoop duct length;

(b) By making the scoop entrance project definitely forward out of the somewhat conical cowling contour; and

(c) By putting a "dash pot" in the aneroid (though this alone did not cure one minor detail of metering disturbance).

As haste was urgent, no pressure records were taken.

There is reason to believe, however, that resonance pressure waves in the scoop can upset carburetor metering if they happen to coincide with the resonance frequency of metering air chambers in the carburetor; in fact, we have had a definite experience like this in a float-type carburetor, details of which I am not at liberty to reveal here. The basic laws of pressure waves in pipes are fairly well known,



■ Fig. 32 – Distribution of pressures along tube in Fig. 30 with blast interrupted in resonance frequency of tube

but an ocular demonstration of their magnitude was shown in the test illustrated in Figs. 30, 31, and 32. As part of the previously mentioned investigation, we set up a tube about $5\frac{1}{2}$ ft long with manometer pressure heads at regular intervals. A blast of air was directed toward the mouth of the tube and interrupted by a rotating disc driven by an electric motor. Fig. 30 shows the set-up and Fig. 31 the shift of the manometers under the steady jet of air uninterrupted, while Fig. 32 shows the variation in pressure distribution with the interrupter disc in rotation. I might note that we found it rather difficult to excite definite waves of fundamental frequency in the pipe by this method, a fact which has also been reported by other experimenters.

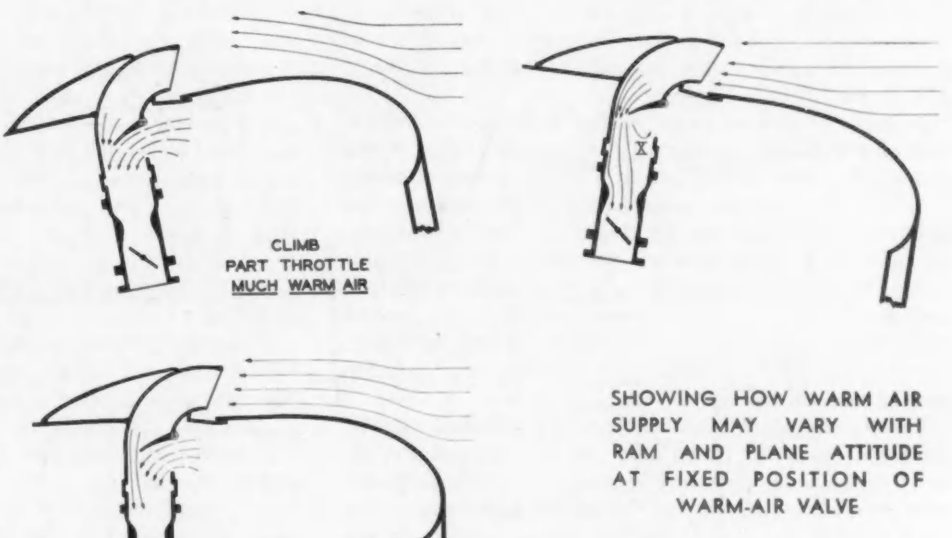
As stated in the foregoing, this difficulty has been cured by relatively simple means, which are now available. We are inclined to believe, however, that formation of pressure waves due to air speed probably depends a good bit upon the exact formation of the air opening with reference to cowl curvature, and direction of air flow. It is common observation that, to excite a definite frequency note in a bottle, tube, or musical wind instrument without a reed, requires that the air blast be directed in just a certain way with reference to the orifice. Apparently the exciting blast must be relatively neutral with reference to the air surge in and out so that it can reinforce either one, once the vibration is set up. It is conceivable that a slight change in the air opening say, for instance, an overhanging top extension forward, might prevent excitation under steady air flow over the cowling. This again is a subject for wind-tunnel or other aerodynamic investigation.

In general, it is difficult to conceive of a ready method of detecting pressure waves of characteristics which will disturb air metering, without the use of some such indicator as the one previously described. An analogy suggested by Figs. 31 and 32 would be to place the several pressure connections along the scoop duct, connect them to respective air-speed gages, balanced on air speed static, and look for a change in their gradation of readings. Fortunately, cases of disturbance from this cause have been recorded rather seldom.

■ Warm Air Valve Systems

At the present time it is current practice to fit to the scoop an alternative warm air passage, with selecting valve, for use when there is heavy atmospheric precipitation or when atmospheric conditions favor formation of ice in the intake system. It is well known that many of these warm air controls are generally unsatisfactory, but the reasons therefor are less generally understood. The current types are particularly bad when, as recently, a large heating muff is supplied so that a graduation or mixture of warm and cold air is necessary to protect against ice on the one hand, and to avoid overheating on the other. The usual valve construction is objectionable for the following reasons:

1. Due to the fact that the cold air is under a variable



■ Fig. 33 – Disturbance of warm air regulation by plane attitude and throttle position, with usual arrangement of warm and cold air intakes

degree of ram according to the ship attitude, while the warm air is not, for a given intermediate position of the heat valve, a greater proportion of cold air will come in in level flight or in a dive than during climb when most needed. (See Fig. 33.) It is a common experience for a thermocouple located at X to read cold-air temperature at level flight.

2. Similarly, for a given percentage opening of the valve, more warm air will be drawn through in proportion to the cold air at large throttle openings than at small throttle openings: at small throttle openings enough cold air comes through under the ram differential to satisfy the carburetor demand. This, like (1), is exactly opposite to what the engine requires. These two faults are inherent in any type of scoop where the cold air is under ram and the warm air is not.

3. It is impossible to shut off entrance of heavy rain or thick wet snow without getting full hot air, which may bring the intake temperature too high for engine safety.

4. With such a construction the cold and warm air do not mix well, but pursue separate paths down through the carburetor and adapter. This condition does not give good protection against ice formation and makes it necessary to use a hotter air intake charge than otherwise would be necessary. In addition, it is impossible to locate a thermometer bulb in the system and have it give consistent readings that will indicate the minimum temperature that is safe from ice formation.

In addition, the ordinary shape of warm air valve is objectionable in the full-hot air position in that the air passage entrance is abrupt across the carburetor and does not give good entrance air flow. Fig. 34 shows the pressure distribution of a common type of scoop in the full-warm-air position.

All the deficiencies just enumerated disappear with the construction shown in Fig. 35 in which there are two valve systems and three air supplies: cold outside air; warm "protected" air; and hot air. The warm and hot air supplies are subject to the same ram or lack of ram. As shown in sequence; first position is full cold air; second is with outside air fully shut off and protected air from back of the cylinder baffles, usually at a temperature of about 40 F above outside air; third is part warm and part hot air, as required by weather conditions and the pilot's judgment; and fourth is full hot air.

Attention is directed to the relation of the forward extension of the island member relative to the center of the warm air entrance. This insures a balanced flow of warm air down the front and rear halves of the scoop to the carburetor and a considerable degree of mixing. The particular shape of scoop shown is not perfect as regards uniformity of pressure across the carburetor entrance, but it is very good and this type of scoop in service has proved quite satisfactory. The scoop structure shown in Fig. 1 has similiar airflow characteristics to this and probably would be as satisfactory.

There is one point that should not be overlooked, namely our definition of "protected" air. All the rain and precipitation that hit the front cowl opening around the propeller must eventually go back past the engine baffles and it can easily happen that one of the streams so created should flow into the warm or "protected" air opening unless special means are employed to prevent it. The same could conceivably happen with reference to the intake air entrance on the hot air-muff. Such a wash is improbable and would no doubt be apt to leave evidence of its passage over the otherwise dust-covered surfaces, but anyway it is not altogether safe to assume that air from behind the baffles is entirely free from precipitation.

Conclusions

1. Scoop difficulties occur less frequently than is generally believed. The airscoop has repeatedly been blamed for such diverse ailments as supercharger surge, poor manifold distribution, vapor lock, and many other deficiencies for which it is similarly not guilty. Nevertheless there remain certain problems which, though we can diagnose them, we do not fully understand.

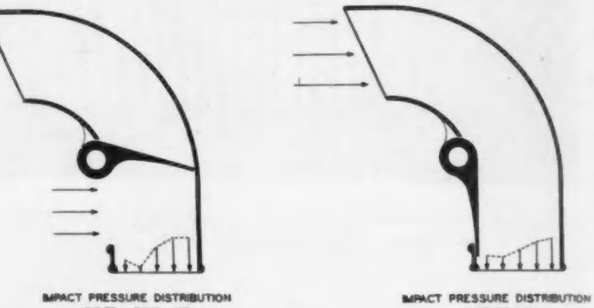


Fig. 34 – Irregularity of pressure distribution with cold-air intake close to carburetor entrance

2. The most important disturbance of metering in flight is enrichment at altitude and high power. This disturbance may be connected with scoop form and propeller slipstream effect. While the difficulty may be partly cured by change of carburetor setting, it is preferable to try to restore normal air-metering conditions in the carburetor by flight research to find the cause. In this work, measurement of air flow (as closely as possible) and of carburetor metering suction are of most direct guidance.

3. There are occasional cases of disturbance from pressure pulsations in the intake duct. We believe that these can always be dealt with as here outlined.

4. Since many of these difficulties occur only in flight and at present cannot be reproduced in ground laboratories, ***they must be studied in flight.*** Such work is obviously the combined function of the ***external air-flow*** expert, or aircraft engineer, and the ***internal air-flow*** expert, or carburetor engineer.

5. It is quite important for the time being that we abandon the conception that carburetor settings must be determined *only* by ground stand test, and amend our procedure accordingly. What is important is that the engine should have its required fuel-air ratio in flight; not that it should have the same jet size as will produce this ratio on the ground, but a different ratio in flight. Eventually, however, we should increase our collective knowledge to a point where we can regularly determine a flight setting from ground test, and where check flights will be necessary only to confirm that the ground fuel-air ratios are reproduced.

6. In both military and commercial procurement of *new* planes, procedure should regularly provide for prototype test of all engine accessories whose performance may be

affected by vibration, temperature, or air-impact conditions, provided that these conditions are different in the new installation from previous practice.

7. In the carburetor it would seem that:

(a) Multiple air-speed elements are preferable.

(b) The impact and velocity members should be as close together as possible in order to have the same position with reference to "nodes" of pulsation in the scoop. Also the venting area of any air-pressure chambers should be small with reference to the volume in the carburetor, so that their resonance frequency will be lower than any frequency in the scoop.

(c) Elastic pneumatic members, such as aneroids, should be damped for both low- and high-frequency vibration.

8. As to airscoops:

(a) It now seems that an elbow shape that gives a uniform pressure pattern at the carburetor face will give less metering disturbance, and is likely to give greater "ram" to the engine. For new designs this should be experimentally checked in prototype flight.

(b) If, for basic design reasons, the scoop entrance must be brought close to the propeller, it is probably safer to give the opening a wide "circumferential aspect," about four times the width of the propeller blades at that radius.

9. As to warm air supply, since vanes may definitely be needed in the scoop elbow, warm (protected) air and hot air should be supplied between the vanes and the scoop entrance, to avoid icing trouble. Proper air temperature control involves that:

(a) The proportion of cold and warm air should not be affected by ram differential.

(b) The control valve should not affect the carburetor metering.

(c) The warm and hot air should be mixed; and the hot-air stream should not localize on the aneroid.

(d) It should be possible to close completely the cold-air entrance without getting "full-hot" air.

■ Acknowledgment

In the work here reported we wish to acknowledge the help of Pratt & Whitney Aircraft, of Wright Aeronautical Corp., of North American Aviation, and of other aircraft companies and several of our commercial airlines. Much of our own work should be credited to M. R. Balis, E. J. Partington, and S. B. Smith of our experimental departments.

DISCUSSION

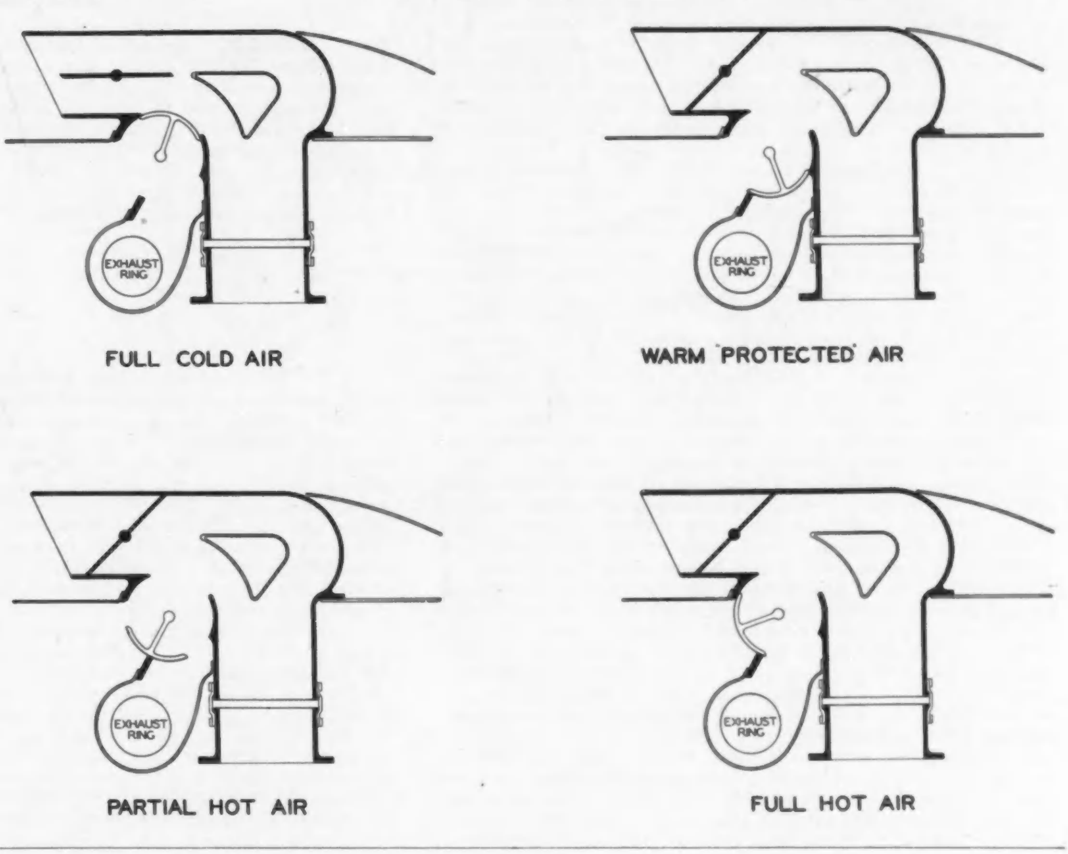
Effect of Aneroid and Carburetor Metering Elements

-R. C. Palmer

Project Engineer, Pratt & Whitney Aircraft, Division of United Aircraft Corp.

THE following discussion is published in the interest of presenting certain considerations not covered in Mr. Mock's paper. For purposes of brevity, discussion is confined to the conclusions which summarize his paper. The succeeding paragraphs are identified numerically to agree with the numbering of Mr. Mock's conclusions.

1. In general, it is agreed that scoop difficulties occur less fre-



■ Fig. 35 – Improved form of airscoop with three-position valve system

quently than is generally believed, but it is definitely true that scoop designs can influence manifold distribution to a marked degree. If, for example, a given scoop handles the air flow in such a manner that a high-velocity air jet exists at the discharge flange either at the rear or the front of the scoop, not anything in the functioning of the carburetor venturi will correct this localized high-velocity condition as the air passes through them, and hence, the air velocity distribution at the impeller eye is also disturbed. In the design of supercharger passages, the application of straightening vanes and the determination of an optimum impeller eye diameter is usually predicated on a uniform velocity pattern across the discharge side of the carburetor.

A specific example of poor manifold distribution was recently encountered in service and corrected by employing turning vanes in the scoop elbow so that velocities across the discharge face of the scoop would be more uniform.

2. During the past year and a half it has been continually necessary to correct carburetor and scoop combinations in the field because of excessive enrichment at altitude, particularly at the higher power outputs. Indeed, the condition is so chronic as to exclude a single installation of the single-stage two-speed engine where this trouble does not exist. Mr. Mock, in his paper, apparently is attributing all of this malfunctioning to the scoop and has almost completely neglected to attribute some of the causes for enrichment to the carburetor metering elements themselves. It would surely seem that of the dozens and dozens of scoops tested throughout the country during the past year, at least one would have a satisfactory influence on the carburetor at altitude. Results of extensive flight tests, ground engine tests, and airbox tests at the Navy's Aircraft Engine Laboratory are pointing more and more to the inadequacy of the present aneroid to correct for density both from the viewpoint of temperature and for pressure. The currently used aneroid is a typical capsule, nitrogen-filled, which supposedly gives both pressure and temperature compensation. It now appears that corrections for both temperature and pressure can be accomplished only by using a separate instrument for each function. Lack of temperature compensation in the current aneroid is accounting for some part of the enrichment now experienced at altitude and high powers. Further, all the data point to the fact that the proportions of the metering elements must be changed in such a manner that velocities through the venturi will be reduced to allow a maximum corrected metering suction of 50 in. H_2O. By so doing (refer to Fig. 28) carburetors will operate in the range of nominal venturi velocity which will permit their functioning along a true parabola. Our present practice is to operate in the neighborhood of 80 or 90 and 100 in. H_2O for take-off air flow. For some time now, we have been working on the premise that departures at high air flows at high altitudes from the theoretically correct velocity-squared relation is directly attributable to scoops. However, the effect of reduced temperatures at altitude is undoubtedly precipitating the functioning of the venturi toward a so-called critical condition irrespective of the scoop action. It is for this reason that the metering element proportions must be changed so as to put actual operating velocities in the non-critical range (below 300 fps referring to Fig. 28).

Lacking concrete proof of the malfunctioning of automatic carburetors due to scoops, it appears that this paper has directed too little attention toward the carburetor itself.

3. This paper deals at length with the adverse effect of pressure waves in the scoop induced by propeller blades passing in front of the scoop opening. While it is true that such an action can produce an effect on metering, it is nevertheless felt that it is not of too great importance in the analysis of enrichment at altitude and high powers. Mr. Mock in this paper indicates that no case of malfunctioning in carburetion *in flight* has ever been attributed to pulsations or pressure waves in the airscoop. He does state that probably some ill effects would be felt in reving-up on the ground or at take-off. It is interesting to note that only isolated cases of poor carburetion at take-off have been reported and it is felt that, if a bad condition did exist, it could be very readily corrected. Discussing reasons for the change of air metering characteristic, Mr. Mock states: "Periodic variation of flow rate: since the depression varies as the momentary velocity squared, the integrated depression will be greater in pulsating than a steady flow for the same average mass flow per unit time." This relation is unquestionably true but, in analyzing a specific case of carburetor metering in flight, if we assume that two conditions might exist, firstly, where the velocity in the scoop is oscillating between 100 and 200 mph (an extreme case) and, secondly, a constant velocity of 150 mph for the same mass air flow, the net effect on the fuel flow from the carburetor will be only 3.9% difference for the two conditions. This is a relatively small percentage enrichment when we consider the very wide discrepancies which are being experienced with our present carburetors.

4. During the various ground studies we have neglected to include one fundamental variable, namely, temperature. All of the work in airboxes and on the test stands, with the single exception of tests at the Navy Department's Aircraft Engine Laboratory, has been at relatively high air temperatures, at or above standard conditions for sea level. It would seem most important that any ground studies of scoop performance, venturi characteristics, and altitude compensation should be made at standard altitude temperatures. Probably in this connection some of the larger government-controlled laboratories where a range of temperatures is available could be of great assistance in these problems.

5. The fifth conclusion is a dangerous one and one which will provide no end of trouble. Aside from the terrific amount of work which will be involved in establishing carburetor settings in flight, probably the two most valid objections to this procedure are: (a) different carburetor settings for each model of airplane; (b) non-interchangeability of carburetors on the same engine model. It is of paramount importance, of course, under present circumstances, that any given group of engines of the same model should be such that they can be readily transferred to another airplane installation *without change*. It is recognized, of course, that under certain conditions of urgent delivery requirements establishment of some carburetor settings in flight is unavoidable, but it is strongly believed that each case of this sort must be regarded as an extreme evil and accepted only after every resource has been exhausted. In other words, the carburetor which requires change for each installation is neither a practical nor an acceptable instrument.

9. As a result of extensive work by the Carburetor and Icing Study Groups of Pratt and Whitney, it is now possible to cope satisfactorily with the problem of providing hot air without getting into icing trouble or adversely affecting the metering. A number of excellent scoop designs have recently been put into service by various airplane manufacturers and airline operators, all of which should be trouble-free as regards carburetor metering. Pratt and Whitney has designed, bench-tested, and flight-tested a scoop incorporating a hot-air door which seems to be quite satisfactory.

In general, the application of a correct hot-air door to a scoop is purely a mechanical consideration which heretofore has been given too little thought and little or no testing. As stated previously, an intelligent approach to the problem by different persons has resulted in several excellent arrangements.

Solution of Difficult Carburetor Problem

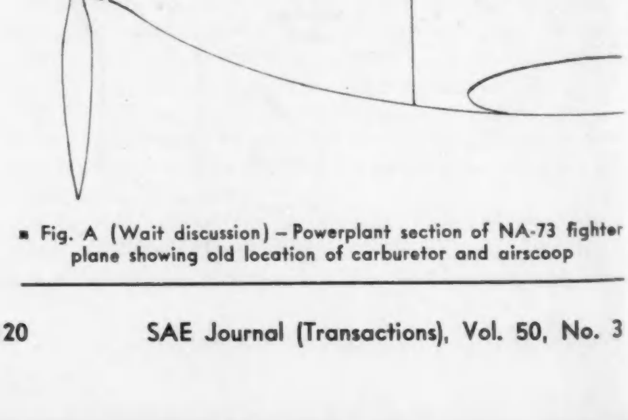
– L. S. Wait

North American Aviation, Inc.

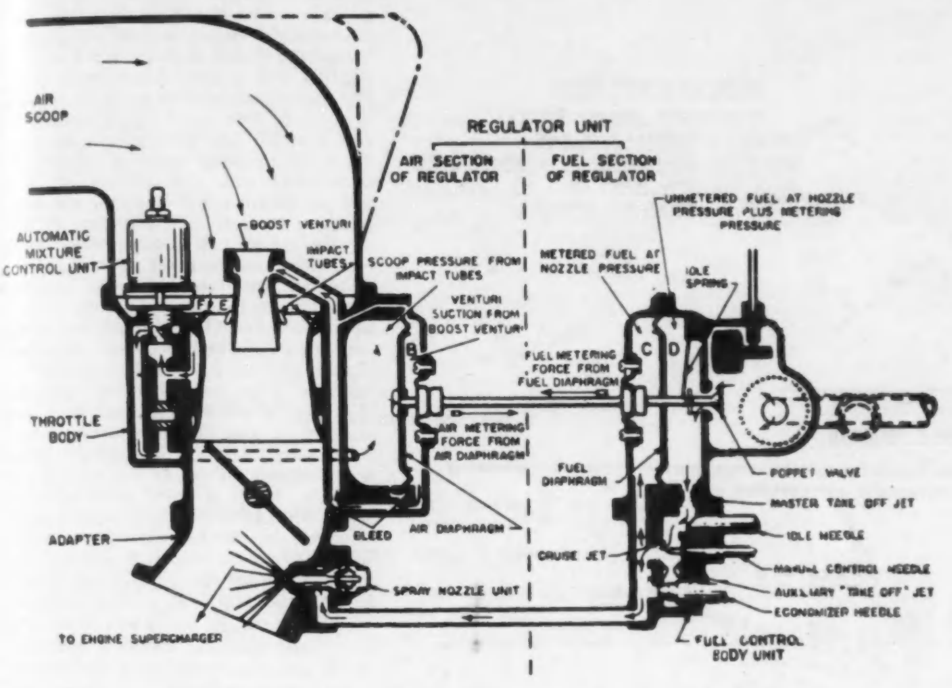
THIS discussion has been prepared by members of the Power Plant and Flight Research sections of the Engineering Department of North American Aviation, Inc. Its purpose is to present, in brief form, the methods used to determine in flight the cause and subsequent correction of a difficult carburetor problem.

As will be shown, the problem can be classified as pressure-wave effect, which has been mentioned by Mr. Mock in his paper.

Fig. A is an outline of the powerplant section of the NA-73 fighter airplane, showing principally the location of the carburetor air intake

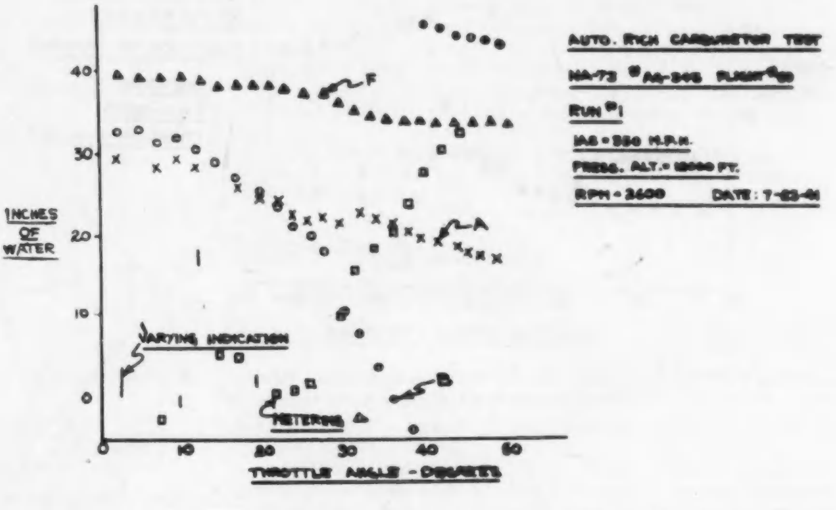


▪ Fig. A (Wait discussion) – Powerplant section of NA-73 fighter plane showing old location of carburetor and airscoop



■ Fig. B (Wait discussion) – Cross-section of Stromberg injection carburetor

■ Fig. C, below (Wait discussion) – Carburetor pressures versus throttle angle obtained in steady-flight condition

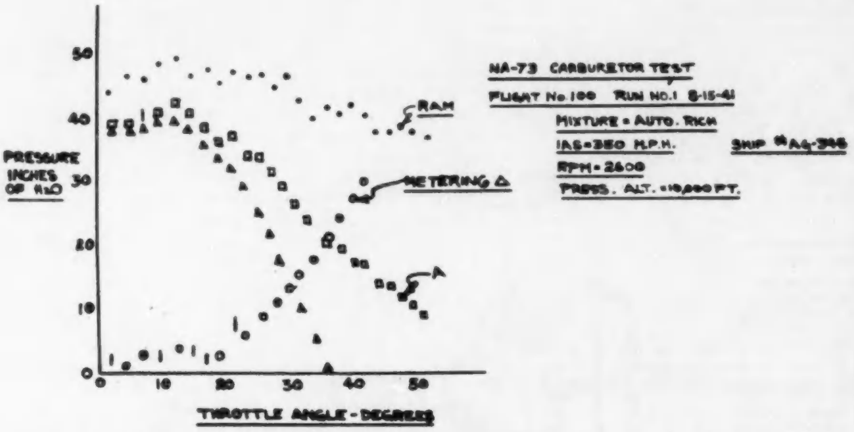


scoop relative to the cowling, propeller, and the carburetor. It is approximately to scale. It should be pointed out that this installation gave excellent metering characteristics in all conditions of flight except in a dive at part throttle. The erratic engine operation in a dive would occur at any rpm, any airspeed above approximately 300 mph and at any throttle opening less than 30 deg. Slight variations in the characteristic of the trouble could be obtained with various mixture strengths and so on; however, the basic condition can be described as follows:

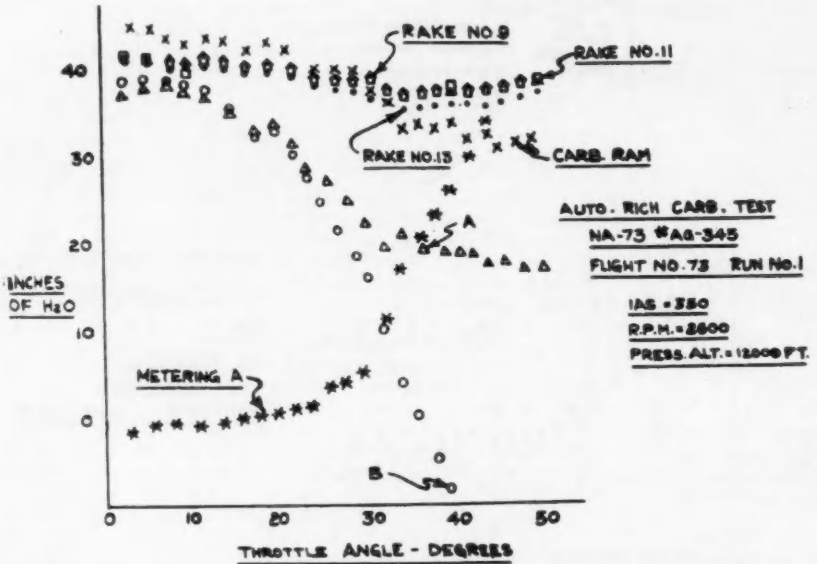
With the throttle approximately ¾ open, the pilot would nose the airplane down into a gentle dive and, as the speed increased, the natural reaction of the pilot would be to retard the throttle. At the higher throttle angles, the engine operated smoothly, of course, but, as the throttle was closed further, the engine would quit. Further motion of the throttle between approximately 25 deg and fully closed would have no effect upon the engine whatsoever. Then, as the procedure was reversed, that is if the pilot would open the throttle during the same dive condition, the engine would be inoperative until the critical angle was exceeded – at which time the engine would take hold suddenly, developing approximately 600 hp and, as you can readily believe, the reaction on the airplane was quite great. The condition also was found to be extremely dangerous as, on many occasions as the throttle was opened and the mixture strength increased, the engine would sometimes start and, at others, would develop a terrific backfire sufficient in intensity noticeably to deform the air intake scoop. Also associated with the engine roughness and stoppage in the dive was the fact that the top cowling, principally that section just ahead of the carburetor elbow, would vibrate quite visibly – with vibration of such intensity that the exciting force must be of large proportions.

To analyze properly the effects on engine mixture strength that were taking place, we will look at a cross-sectional diagram of the carburetor, (Fig. B). This analysis deals only with the left-hand section, or air section of the carburetor regulator unit. Included in this section are principally the airscoop, the impact tubes, boost venturi, automatic altitude compensator unit, the air diaphragm which separates Chambers *A* and *B*, and the throttle. The operation of the Stromberg injection carburetor is well known and its operation will not be described only to mention that the fuel supplied to the engine through the spray nozzle unit is controlled by the differential pressure across Chambers *A* and *B*, such differential being derived from the air flow through the scoop directed against the impact tubes, which supplies pressure to Chamber *A*, and the suction created in the boost venturi which lowers the pressures in Chamber *B*. Then, as the air flow increases or decreases, as the throttle is opened or closed, the air diaphragm is caused to move which, in turn, feeds more or less fuel through the spray nozzle unit, always maintaining the desired fuel-air ratio. To visualize properly the functioning of the air regulator in flight, sensitive pressure instruments were connected to Chambers *A* and *B* to obtain the metering differential across the air diaphragm, and in this case pressures were also taken at the top carburetor flange to give carburetor ram and at *F*, the pressure behind the venturi.

It can also be seen that, as the plunger of the automatic mixture control unit moves downward, the pressure in Chamber *A* decreases, lowering the metering differential, thereby creating a leaner mixture. The following diagrams show the results obtained in flight where the measured pressures were impact pressure as measured in Chamber *A*, venturi suction as measured in Chamber *B*, the metering differential pressure or difference between *A* and *B*, the carburetor ram as measured at the carburetor top flange, the impact pressure or pressure



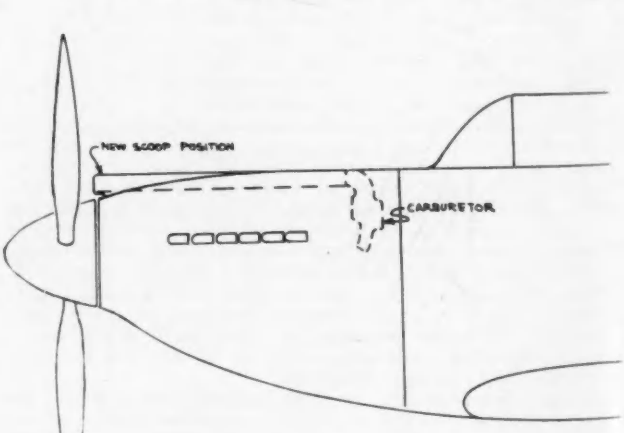
■ **Fig. D (Wait discussion) – Carburetor pressures versus throttle angle after damping the automatic compensator unit**



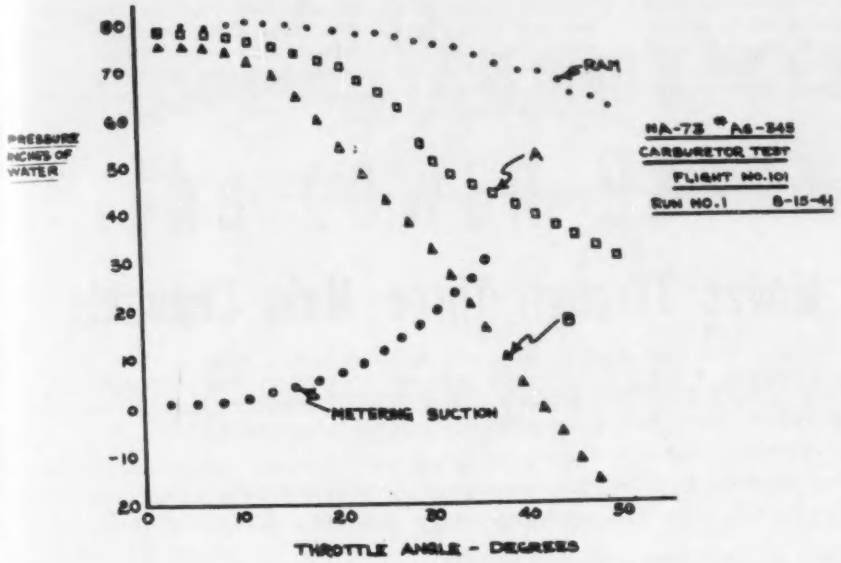
■ **Fig. E (Wait discussion) – Carburetor pressures versus throttle angle after installing oil-dashpot-damped altitude compensator unit**

behind the venturi, shown as *F* and, in some cases, the total head pressures obtained on a rake mounted just above the carburetor. Fig. C represents various pressures in the carburetor plotted against throttle angle in degrees obtained in a steady-flight condition, where the only variable is throttle angle. In this instance the indicated airspeed was held constant at 350 mph, the rpm was constant at 2600, the approximate pressure altitude was 12,000 ft, and the throttle angle was decreased from approximately 45 deg to fully closed, the readings being taken photographically approximately each 2 deg of throttle movement. It should be mentioned that these data are all obtained during a period of approximately 15 sec. This diagram represents the conditions obtained with the original carburetor scoop and with the standard altitude compensator as furnished with the carburetor. It can be seen that the impact pressure is quite steady that, for the larger throttle angles, the pressure in Chamber *A* is quite steady; the pressure in Chamber *B* is consistently lower and quite steady; and the metering differential is quite steady. It will also be noted that, down to 30 deg, the upper portion of the metering differential curve maintained its characteristic parabolic shape. However, below 30 deg throttle angle, the metering differential takes a nose dive, dropping almost to zero and from there on down is extremely erratic. It also can be noted that the pressure in Chamber *A* drops below the pressure in Chamber *B* giving a negative pressure differential. From this curve, it is easy to believe that the engine would stop. Our first analysis of this condition showed that the venturi suction as represented by Chamber *B* maintained its characteristic parabolic shape, whereas the pressure in Chamber *A* became erratic and its tendency was to drop as the pressure behind the venturi tended to increase. (See Fig. B.) This result led us to believe that the altitude compensator must be at fault as the only difference between pressures *F* and *A* is that caused by the pressure drop across the metering orifice of the altitude compensator unit. Consequently, our next step was an attempt to damp the automatic compensator unit. This was done by filling the control unit housing with a light oil, totally immersing the sylphon bellows in oil but still maintaining the vents at the top of the control unit, so that altitude compensation could be obtained and at the same time any high-frequency vibrations of the control unit bellows would be damped. This gave improved operation (Fig. D). It will be seen that, under the same flight conditions, metering differential came closer to a parabolic curve and, although the engine still ran slightly rough, it would not stop completely and the backfire conditions no longer existed. However, since the measured pressures still showed erratic tendencies, a pressure rake was installed just above the carburetor and three of the total head readings obtained on this same flight are shown here. The three rake readings, although taken at widely separated points across the airscoop section, were reasonably consistent and steady and it was extremely hard to believe that any stratification or other air turbulence, having sufficient strength to completely upset the metering characteristics, could exist.

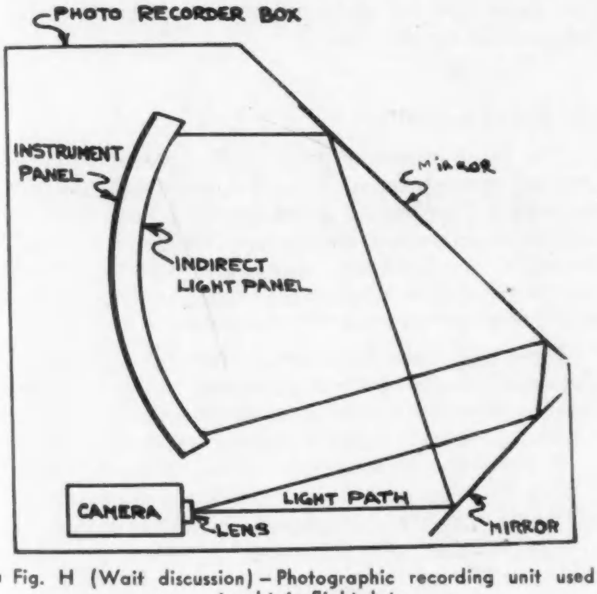
At this time many other minor changes were made in an effort to improve further the engine operation. Two of these were modifications in the scoop elbow immediately above the carburetor (Fig. B). The first dotted line shows how the elbow was squared in an effort to improve the flow. The broken line to the rear shows a second elbow change that was made. Neither of these modifications altered the air flow or engine operation characteristics as could be determined by our measurements. Being pressed as always for airplane deliveries, it was decided that a previously designed oil-dashpot-damped altitude compensator unit must be obtained for installation on the airplane. This was installed and when



■ **Fig. F (Wait discussion) – Powerplant section of NA-73 fighter plane showing new location airscoop**



■ Fig. G (Wait discussion) – Carburetor pressures versus throttle angle after lengthening the airscoop



■ Fig. H (Wait discussion) – Photographic recording unit used to obtain flight data

flown showed the characteristics shown in Fig. E. The metering differential is shown to be quite steady through the upper portion and only slightly variable at the lower end. Also the engine operation was considered acceptable. It should be pointed out that the ram pressure was always higher than either the boost venturi or impact pressure even with the throttle completely closed, where theoretically no air is flowing through the throttle body and all pressures should be identical. At this time an analysis of the airscoop length demonstrated conclusively that its normal air vibration frequency of 50 cycles coincided exactly with the normal frequency of the altitude compensator unit and only an extreme shortening of the scoop could alter this situation. (See Fig. A.) At this time a test was made with the scoop shortened approximately 12 in. without any change in characteristics. Also a short scoop was made that projected above the carburetor, the lower lip being approximately 2 in. above the cowl contour. This change did fix the dive condition; however, the metering characteristics in climb and level flight were altered with the engine running considerably richer than previously, and the carburetor ram being lower. Also this configuration could not possibly have been used because of a direct interference with the pilot's vision and gun sighting arrangement.

Still not being satisfied with the engine operation, a final modification was tried. (See Fig. F.) This consisted of lengthening the scoop to overlap the spinner slightly, giving a clearance of about 1 in. to clear the boundary layer. When this modification was made, it was known that the normal vibrating frequency of the scoop would be lowered only slightly and no benefit could be expected therefrom. Naturally, we were quite surprised to find that in flight the engine operated smoothly in all dive conditions, the previously mentioned visible vibration of the top cowl had ceased entirely and with the standard undamped altitude compensator, both power compensation and altitude compensation were perfect under all conditions of flight. (See Fig. G.) After scanning pages and pages of erratic diagrams, it was a pleasure to look at the results of this flight where the metering differential maintained a perfect parabolic shape, the ram pressure was steady as was the pressure in Chambers *A* and *B* and further, as the throttle was fully closed, the points met as they should. It will also be noted that the ram was increased considerably which gave a higher critical altitude and consequently a substantial increase in top speed, although the profile drag of the new scoop was slightly greater. Needless to say this is the scoop now used on the production fighter airplane.

It may be of interest to show in Fig. H the photographic recording unit that was used to obtain the flight data herein presented. Eight standard aircraft instruments are mounted on an indirectly lighted instrument panel which is photographed through two mirrors. This forms a self-contained unit, the dimensions of which are approximately 16 x 16 x 9 in. high. These small dimensions are necessary because of the limited space available on a pursuit-type airplane. The camera is a standard 16-mm magazine-loading type, and all photographs are taken by operating a switch mounted on the airplane control stick.

It is the sincere hope of North American Aviation that the results of these tests may be helpful to other manufacturers in correcting similar difficulties.

Author's Closure Elaborates on Temperature Correction

– Frank C. Mock

Bendix Products Division, Bendix Aviation Corp.

As to temperature correction discussed by Mr. Palmer, we have the following three points:

1. As Mr. Palmer states, a simple gas-filled aneroid gives exact temperature correction only with external pressures near that of its filling, and meters indicate rich with decrease of temperature at altitude. We have found, however, that, since there is not much air-temperature spread at high altitude, a fair approximation often can be made by fitting the aneroid calibration to the standard mutual pressure-temperature curve.

2. Due to difficulties of engine cooling at altitude, a uniform fuel-air ratio for constant power at increasing altitude is not always safe. Apparently proper values can be determined only by flight test.

3. In many past installations the aneroid did not receive the average temperature of the entering air. Requoting Conclusion 9 (c) of my paper: "The warm and hot air should be mixed, and the hot-air stream should not localize on the aneroid," we should add that the aneroid should not receive local radiation or conduction from the engine. Again, this can scarcely be checked except by flight test.

In view of the foregoing, we recommend that thermocouples across the air scoop, and one on the gas chamber of the aneroid, be used on every carburetor flight test.

As regards venturi velocity limitations, a decrease of 20% in absolute temperature will decrease the critical velocity 10%, but increase the air density 20%, so that the mass air flow rate permissible increases with decrease in temperature. Mr. Palmer's recommendation of 50 in. H_2O at sea level corresponds about to 400 fps nominal velocity at 18,000 ft altitude for the same mass air flow. We now believe that this value can be increased about 12% without undue enrichment.

It is believed that Mr. Palmer's comment does not really conflict with our Conclusion 5, except that we emphasize the need for flight research and occasional check before the status desired can be reached. No such complex engineering development as this can be achieved without experiment under actual or fully reproduced working conditions, and it is in this last detail that our present carburetor procedure falls short.

In regard to Mr. Wait's discussion, I would like to say that this is a typical example of the difficulties with which we have to deal, as well as of the approach necessary for a satisfactory solution. Once we realize the existence of phenomena such as those described, we soon learn how to deal with them and they are no longer a problem. The technique and equipment used will be particularly helpful if and when other similar problems arise.

Rubber Research Moves Through Three Main Channels

THERE have been three major channels through which research has moved to make possible the use of rubber products as we know them today.

1. Vulcanization

Most laymen know that the first step toward using rubber in a practical manner was the discovery of vulcanization through the accident of dropping a rubber and sulfur mixture on a hot stove, but the layman does not know just what phases of this accident were important. The most important was temperature, and the happy accident of this factor being just right at the instant of the mixture's arrival on the hot stove. If the stove had been hotter, the rubber would have been destroyed and, if cooler, the resultant vulcanization would not have taken place.

This truth is still one of the fundamentals of the rubber industry.

The temperature and length of time to which any rubber product is subjected to this temperature, have not only a marked effect on the quality of the finished product, but are calculated especially to make that product do the best job in the service for which it was designed.

In calculating this time and temperature, the size and shape must be considered. Roughly, the larger the object, the longer the time necessary to cure. However, the basic factor in these calculations is the compound. Rubber chemists prepare their stocks much the same as steel chemists prepare steel alloys, and for the same reason – to obtain the best material for a given job. This is done by milling into the rubber certain chemical catalysts, or accelerators.

Modern accelerators are the result of tremendous advances in the technology of this part of the rubber scientist's work, results of millions of hours of experimenting, and have increased the strength, elastic properties, and durability of rubber. They also have the practical effect of saving an immense amount of money in increased plant output, through reduction of time of processing and the requiring of less equipment.

2. Carbon Black

The second of the major moves mentioned is the branch of research concerning the use of stiffening or reinforcing powders, which was begun in a purely accidental manner.

The incorporation of finely divided powders into rubber compounds has been common practice in the rubber industry almost since its inception.

In the early years these components were regarded as worth while only because their use entailed replacement of rubber, thus reducing the cost.

Through this time keen investigators noticed that these powders produced hardness disproportionate to the amount used, as well as unexpected toughness.

At this time the most common was zinc oxide; carbon black, while also used in very small quantities, was so used only for coloring. World War I began to make demands on the supply of zinc oxide and the rubber industry at almost the same time was searching for something to improve resistance to abrasion and increase the tire mileage. Experiments determined that carbon black would be the solution to both these problems. The early blacks were borrowed from the paint industry and were not perfectly adapted for use in rubber.

3. Anti-Oxidants

The third improvement in rubber products has been attained through the use of anti-oxidants, or age resistors, in rubber. These are, in general, organic compounds which inhibit oxidation and they increase the life of rubber 300 to 400%. Anti-oxidants increase rubber's resistance to heat and, without them, cracks would appear and rubber would break down at much lower temperature.

These three major developments have not only increased the usefulness of rubber, but have also opened up a multitude of new fields. Together they have had at some time almost the whole chemical world working on problems, each of which in turn has had its influence on the others.

Excerpts from the paper: "New Uses for Rubber," by E. Waldo Stein, The Firestone Tire & Rubber Co., presented at a meeting of the Indiana Section of the Society, Indianapolis, Ind., Oct. 9, 1941.

Error in February Paper on Stress Determination

THROUGH a typographical error the sense of a concluding sentence of the paper: "Facts and Fallacies of Stress Determination," by J. O. Almen, published on p. 61, Transactions Section, February, 1942, issue of the SAE Journal, was reversed completely. As published in error, starting at the ninth line from the end of the paper, this sentence reads: "The stress calculated by such formulas will not be true stress but this is unimportant because we will *not* have reliable means by which to determine load capacity." This sentence should be corrected to read . . . "because we will *now* have reliable means by which to determine load capacity."

Symposium - Commercial Vehicles and LOWER-OCTANE FUELS

TO CONSERVE critical materials for war uses, refiners have been quietly reducing the octane numbers of all commercial gasolines. These lower antiknock values are the result of skimming off the cream, or volatile fractions, for use in aviation gasolines, and of using less tetraethyl lead since practically all components entering into its manufacture – lead, ethyl alcohol, and chlorine – have become critical.

What refiners, engine builders, and fleet operators are doing and recommending to soften the impact of this profound change in commercial fuels, is the subject of this symposium.

■ ■ ■

Part I–From the Refiner's Viewpoint

SERVICE station men throughout the nation may be called upon to serve "gasoline cocktails" to the drivers of trucks and buses essential to the war effort, it is suggested by Mr. Hubner. Owing to "octane stripping" and larger defense requirements of tetraethyl lead supplies, Mr. Hubner says, the quality of regular-grade gasoline will almost certainly decline, but the potential loss in power and performance of most heavy-duty equipment could be made up by mixing a "cocktail" of regular and premium-grade fuels at the service station to meet the needs of the particular equipment.

Although most refiners could meet the deficiency in octane number caused by stripping base stocks of high-octane fractions for aviation fuels by added concentrations of tetraethyl lead, the author emphasizes that allocation of raw materials had limited the supply of this antiknock agent. Present indications, he says, are that tetraethyl lead available for regular-grade gasolines probably will be sufficient only to give such gasolines an average of about 72 octane number. Premium-grade fuels will continue to be produced to supply the needs of the Army, since an 80-octane fuel has been specified for use in all of its mechanized equipment.

by WILLIAM H. HUBNER

Director, Refinery Technology Division, Ethyl Gasoline Corp.

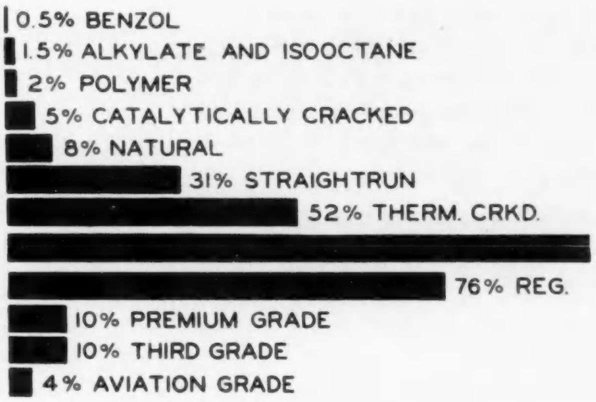
THE aircraft, tanks, and trucks which form the spearhead of our fighting forces depend on the petroleum industry for their fuels and lubricants. Fortunately, both in quantity of petroleum and in quality of petroleum products, the United States is far ahead of its enemies. We have petroleum production and refining capacity to fill all demands. The 100-octane aviation fuel and the 80-octane all-purpose fuel supplied to our Army and Navy, and to our Allies, will furnish the superior performance in the air and on the ground which may prove to be the decisive factor in total victory. But, to supply gasoline and lubricating oils of the quality required and in the quantities foreseen, almost certainly will call for sacrifices by the civilian customers of the petroleum industry, which number between 25 and 30 million people.

In order to produce all the 100-octane aviation fuel possible, petroleum refiners right now are straining their facilities to the limit, and new facilities are being rushed to completion. Maximum production is required not only of aviation base fuel but also of the essential high-octane blending agents: alkylate, iso-octane, and isopentane. In

[This symposium was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 12, 1942.]

many cases, refiners are forced to strip motor fuels of some of their volatile high-octane fractions.

To date, most aviation fuel of 100-octane quality has been manufactured by blending about 40% of 72- to 74-octane straight-run gasoline with 10% of 88 to 90 octane isopentane, 50% of 93 to 97 octane alkylate or commercial iso-octane, and adding 3 cc of tetraethyl lead per gal. However, the limit on lead content recently was raised to 4 cc per gal, so that it is now possible to decrease the vital alkylate content to about 40% which will mean 20% more 100-octane fuel. The straight-run base fuel has been obtained by ordinary distillation of selected crude oils, notably California and Gulf Coast oils. The isopentane has been obtained by fractionation from natural gasoline, the equipment for which is available only in the larger refineries and in certain natural gasoline centers. Likewise, the alkylate and iso-octane are being produced only in the larger refineries, less than 20 in number.



■ Fig. 1 – Gasoline production – estimated 1941

■ War Demands for Aviation Fuel

Now, however, with an estimated demand of at least 150,000 bbl per day for 100-octane aviation fuel, which is three times the 1941 peak production of 50,000 bbl per day, it is obvious that more and more of the smaller refiners, too, will be asked to produce one or more of the several aviation fuel components. It appears likely that most refiners with crude capacities of more than 10,000 bbl per day will be called upon to make alkylate instead of the polymer gasoline now used as a blending agent for their motor fuel. At the same time, they will be asked to strip their motor fuels of all other possible aviation ingredients. This, of course, will include a large part of the catalytically cracked and reformed gasolines of higher octane quality now being used by some of the larger companies in their regular and premium grade fuels. This condition will be true particularly in those sections of the country where high-octane straight-run gasoline is not readily available. It is estimated that there are now some 20 plants producing approximately 100,000 bbl of catalytically cracked gasoline per day, with indications that this figure will be increased to approximately 150,000 bbl during 1942. Except for the small amount manufactured specifically for aviation purposes through the use of a special synthetic catalyst, much of the catalytically cracked gasoline is not suitable *per se* as aviation base fuel; but, by further catalytic treatment, it will yield the desired product – one that is relatively low in unsaturated hydrocarbon content in order to insure good storage stability and a better susceptibility to tetraethyl lead.

In the California and Gulf Coast areas where motor fuels, particularly premium-grade fuels, contain a high percentage of high-octane straight-run, we may expect that this component will find its way into aviation base fuel. Natural and so-called recycle gasolines will be stripped of their highest octane components: isopentane, isobutane and, in many cases, normal butane. The isobutane either will be used at its refinery source as one of the essential materials in the manufacture of alkylate or, if the other essential materials – isobutene and normal butene – are not available, it will be shipped to some plant, possibly a "community" plant, where such materials are available. Some of the normal butane will be converted by isomerization to isobutane. In certain cases, the normal butane will be converted to normal butene either by high-temperature cracking or by catalytic dehydrogenation, in which form it can be used as charging stock for alkylation or as a source of synthetic rubber. Similarly, commercial benzol, which is produced as a byproduct of the steel industry and now used to some extent in motor fuels, will undoubtedly find its way into 100-octane aviation fuels.

Because of this "octane stripping," it is obvious that the base stocks of fuels for civilian use must suffer. How much difference this stripping will make on the octane number of the finished motor fuels will depend largely on the amount of tetraethyl lead available for civilian use. Most refiners could overcome all or nearly all of the deficiency by adding greater concentrations of tetraethyl lead if it were available. Others, already using substantial concentrations of tetraethyl lead, would be forced to lower octane numbers.

■ Leaded Fuels in Wide Use

Tetraethyl lead is used at present in all airline and military aviation fuel and in about 85% of all motor fuel. Approximately 250 refiners in the United States use tetraethyl lead in all or nearly all of their motor and aviation fuels, and, among them, they produce over 90% of the country's gasoline. About 95% of all premium-grade fuels and 85% of all regular-grade fuels contain tetraethyl lead. At the present time, aviation fuels constitute about 4% of the total gasoline produced; premium-grade motor fuels about 10%; regular grades about 76%, and third grades about 10%. See Fig. 1.

The characteristic of gasoline susceptibility to tetraethyl lead makes tetraethyl lead more economical in lower concentrations, and the amount which each refiner uses is determined by the economic balance of gaining the necessary quality by other means. The limit of 3 cc per gal is used frequently by some refiners as a means of producing competitive motor fuels. Tetraethyl lead permits flexibility of operation to the large and small refiner alike. By its use he is able to obtain and maintain uniform antiknock quality in his various grades of gasoline, even though his crude-oil supply or volatility requirements may change, or he may find it necessary to vary the operation of his re-

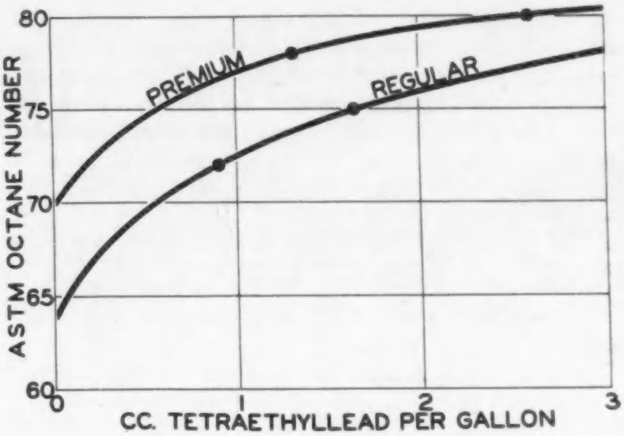
finery equipment in order to produce more or less of any one of the many components used in the manufacture of modern motor fuels.

Effectiveness of Lead

Curves indicating the effectiveness of tetraethyl lead in increasing antiknock value are shown in Fig. 2. These curves are plotted from the average figures for regular and premium-grade fuels during 1941. It should be emphasized that these curves are plotted to pass through the average 1941 lead concentrations of 2.6 cc per gal for 80-octane premium-grade fuel and 1.6 cc for 75-octane regular-grade fuel. The plot indicates that a 40 to 50% reduction in lead content will lower the octane rating of premium grade fuel from 80 to 78 and of regular grade fuel from 75 to 72. The plot also indicates how much more lead would be required to maintain any specific octane level if the antiknock value of the base gasoline for either regular or premium-grade fuel is lowered through necessary changes in blending technique.

In the normal course of events, the strains and stresses imposed upon the petroleum industry by the war would be compensated for by the use of additional quantities of tetraethyl lead. This has been the case during the past few years, as shown by the fact that the demand for tetraethyl lead increased from 128,000,000 lb in 1940 to 160,000,000 lb in 1941. Part of this increase was used to attain the octane levels of regular and premium-grade fuels during that time, and part was used to meet the increased consumption of aviation and motor fuels. Production for the year 1942 has been projected at 198,000,000 lb which is still below the industry's manufacturing capacity. Last summer, as is generally known, the Office of Production Management curtailed the supplies of raw material – lead, chlorine, and alcohol – used in the production of tetraethyl lead. At that time, the Ethyl Gasoline Corp. had a large inventory of finished ethyl fluid. However, by November, 1941, the inventory had been reduced, and it would have been unwise to allow it to be lowered further considering the vital importance of tetraethyl lead from a military standpoint. It became necessary, therefore, for the Ethyl Gasoline Corp. to ration tetraethyl lead to the petroleum industry. "Pearl Harbor" still further emphasized the military importance of maintaining a substantial inventory. The already high estimates for military needs had to be revised upward. The production level has been increased but, even so, it has been found necessary to decrease still further the allocation for civilian needs.

It is apparent from the foregoing that refiners must produce fuels of lower octane number for civilian consumption, partly because of the increased demand for 100-octane aviation fuels and the need for stripping regular and premium-grade fuels of usable aviation components, and partly because adequate supplies of tetraethyl lead are not available. Indications are that the tetraethyl lead available for regular grade gasoline will be sufficient only to give such gasolines an average of about 72 octane number, ASTM Motor method. Already the Ethyl Gasoline Corp. has lowered its minimum octane rating for ethyl gasoline from 80 to 78 ASTM Motor method. A considerable amount of 80-octane fuel will have to be produced to supply the needs of the Army, since an all-purpose fuel of 80 octane number has been specified for use in all of its mechanized ground equipment, but the tetraethyl lead for this purpose is included in the estimated military demands and naturally does not affect civilian allocations. The retention of premium-grade gasoline has apparently been looked upon favorably in many quarters of the government if for no other reason than to permit commercial fleets, which may require a higher octane number gasoline than available regular grade, either to use the premium grade or to blend with this regular grade sufficient of the premium grade to meet specific requirements.



■ Fig. 2 – Average tetraethyl-lead susceptibility – estimated 1941

This analysis assumes, of course, that the civilian consumption of gasoline will remain close to the 1941 level. However, the rationing of rubber tires undoubtedly will tend to curtail civilian gasoline consumption – just how much is difficult to estimate. Although this curtailment will, in turn, tend to relieve the pressure somewhat on the petroleum industry for supplying civilian needs, particularly if the same amount of tetraethyl lead is made available, it will not relieve the pressure so far as the defense needs for aviation fuels are concerned. Less cracking and reforming means less olefinic gases available for manufacturing alkylate. Petroleum refiners could compensate for this shortage by cracking and reforming under more severe conditions which would increase the volume of olefinic gases produced.

As in most other fields, civilians may have to do without the quality to which they have become accustomed but, regardless of the demands, America and its allies can depend upon the petroleum industry to produce the quantity and quality of fuels necessary to win this war.

PARTS II and III of Symposium on following pages

Part II—From the Engine Manufacturer's Viewpoint

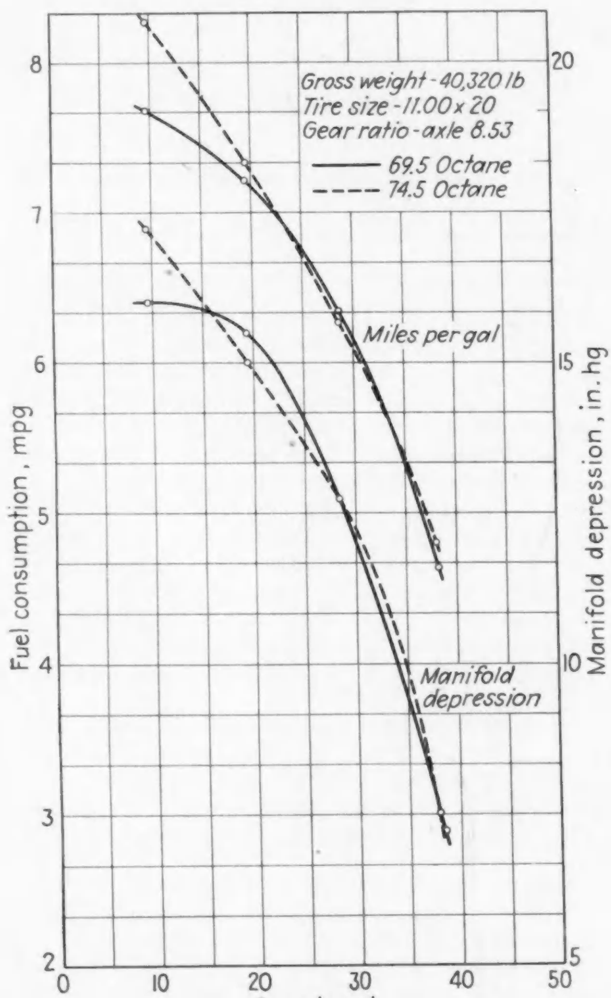
by R. L. WEIDER
Research Engineer, White Motor Co.

As a result of the present world crisis resulting from World War II, we are all very much aware now that, from the problem of supply and demand, certain drastic shortages of materials now exist for regular commercial use. The quality of other materials still available has had to be lowered in some instances, substitutions made in others, resulting in non-consequential effects in some cases, and hardships in others.

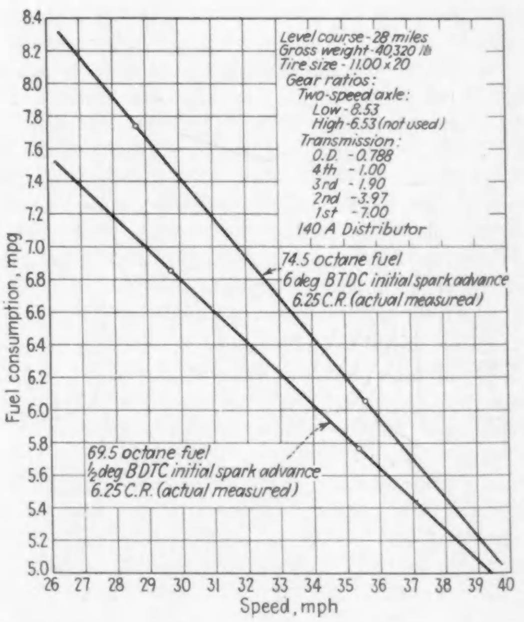
The demand for high-octane gasoline is greater today than ever before in order to keep our aircraft and those of our Allies active and, as the tempo of the action increases, this demand will also rise. Since tetraethyl lead is an important means of securing the high-octane value in modern fuels and, due to other increased demands for the virgin lead and chemical components, the various refiners now find themselves with a restricted amount of tetraethyl lead to treat their gasoline. These restrictions may result in a lowering of the octane number for the standard grade of gasoline from approximately 74 to 68-70 ASTM method.

The trend in engine design in the past has been toward higher compression ratios, requiring higher octane fuels, in order to secure more economical operation. This situation means then that bus and truck operators today are confronted with the question: "What effect will a lower-octane fuel have on the economy and schedule of our operations?" and: "What changes are we going to have to make in order to be able to secure maximum efficiency from our present equipment with lowest maintenance cost?" There are perhaps several approaches to the problem, but we believe answers to the following four questions will serve the operator's best interests:

1. Are we going to have to lower the compression ratio of our equipment?



■ Fig. 1 – White 362 cu in. constant-speed economy road-test results – 74.5 versus 69.5 octane fuels – 6.25:1 actual compression ratio



■ Fig. 2 – Cross-country economy test – 69.5-octane fuel at ½ deg BTDC initial spark advance versus 74.5-octane fuel at 6 deg BTDC spark advance

THE truck and bus operator must know what effect lower-octane fuels will have on his equipment and operation, the author emphasizes. On the assumption that the octane number of standard gasoline may be lowered, from approximately 74 to 68-70, ASTM method, Mr. Weider lists some of the things which an operator of commercial vehicles can expect, using as a basis road and dynamometer tests run with the two fuels on a 362 cu in. White engine used in a 40,000 lb gross vehicle load truck.

1. If an operator has a schedule calling for not over 2400 rpm as a governor setting, he can simply retard his distributor, keeping the compression ratio the same, but he will lose 8% in economy by going to the lower-octane fuel.

2. If an operator has a schedule utilizing 3000 rpm as a governor setting and he simply retards his spark, he will lose approximately 5% in economy.

3. With the high-speed schedule, if he wants to modify his distributor, he can obtain better top end power.

4. With either schedule, if he wants to enrich his carburetor, he can produce the same performance as with the higher-octane fuel if he is willing to sacrifice 10% on economy.

Referring to dynamometer tests made with two compression ratios – 6.28:1 and 5:1, Mr. Weider does not believe that the change to the lower compression ratio should be made. On the other hand, he states that, if an operator does nothing to his units and allows violent detonation to occur, his operating costs are going to be increased by failures of head gaskets, pistons, valves, rings and bearings. Furthermore, cylinder wear will be accelerated, which will further increase both fuel and oil consumption.

Speaking in terms of cost, Mr. Weider summarizes: "On the basis of 100,000 miles per year and with gasoline at a base price of 15¢ per gal, the use of 68-70 octane fuel would cost an operator approximately $150.00 per year per unit more than he is now paying for 74.5-octane fuel."

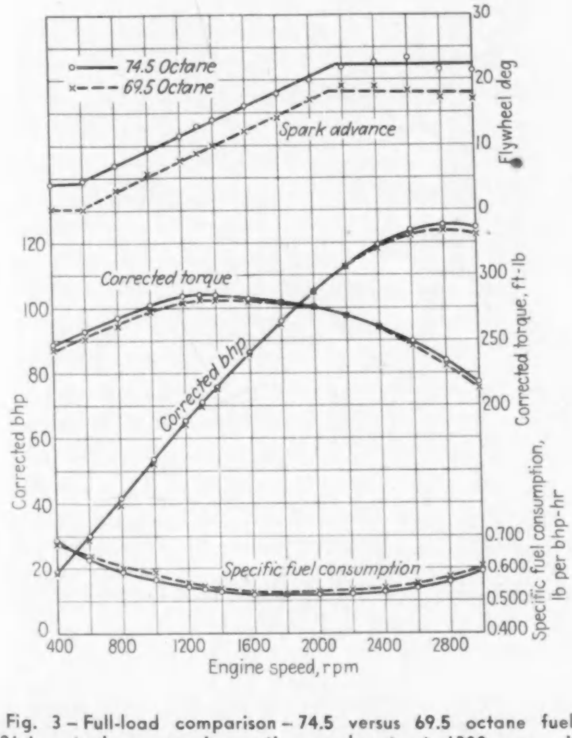
2. Will our present distributor calibrations hold good if we do, or do not, change compression ratios?

3. Will we have to recalibrate our carburetors?

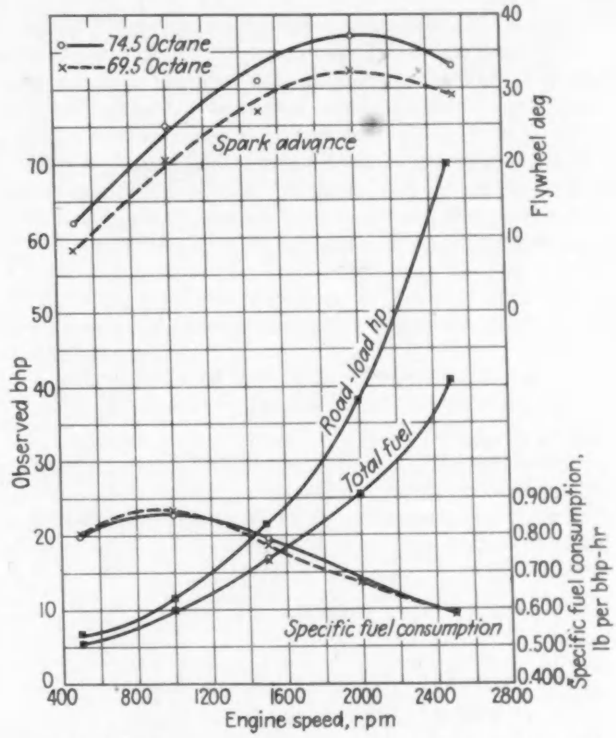
4. With the optimum changes which we can effect, what will be the result on power (or scheduled speed) and economy?

A series of tests, both road and dynamometer, have been run with two fuels on a 362 cu in. White engine as used in our WA-26 model, and the gross vehicle load was established at 40,000 lb, since this load is quite representative of legal requirements. Laboratory inspection data on the two test fuels, 69.5 and 74.5 octane number, respectively, are given in Table 1.

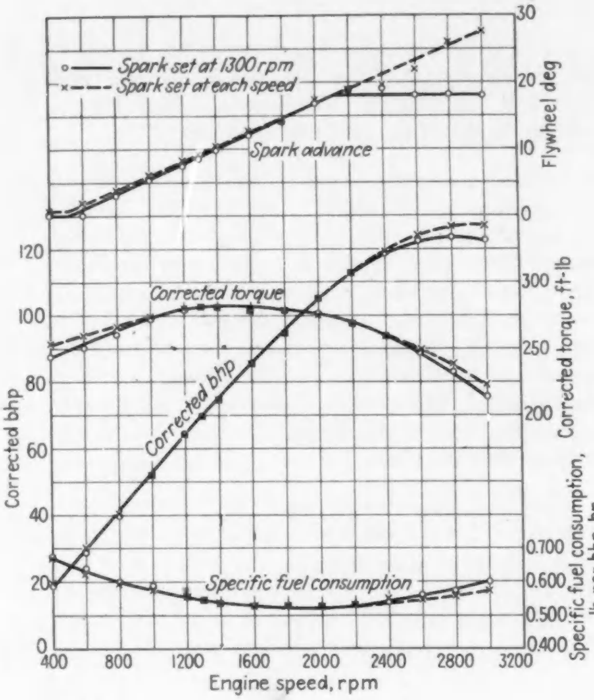
The road-test program consisted of running gasoline



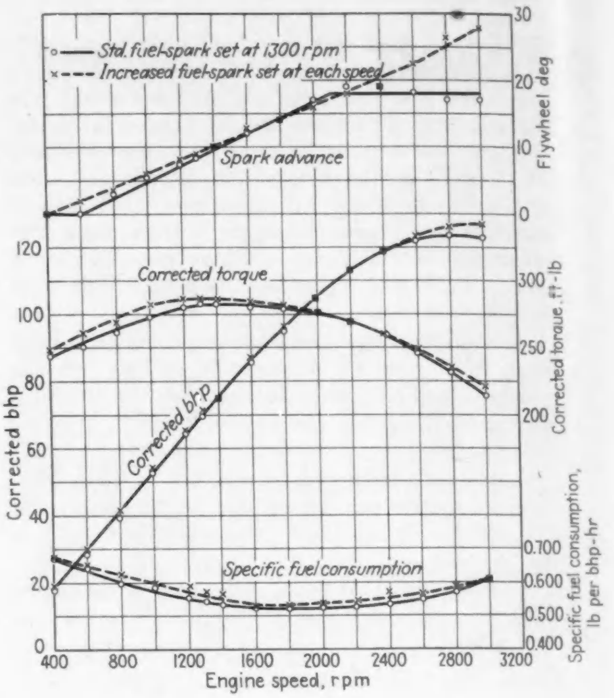
■ Fig. 3 – Full-load comparison – 74.5 versus 69.5 octane fuel – 6.21:1 actual compression ratio – spark set at 1300 rpm with standard distributor



■ Fig. 4 – Road-load comparison – 74.5 versus 69.5 octane fuel – 6.21:1 actual compression ratio – spark set at 1300 rpm with standard distributor



■ Fig. 5 – Full-load comparison – spark set at 1300 rpm versus spark set each speed – 6.21:1 actual compression ratio – 69.5-octane fuel



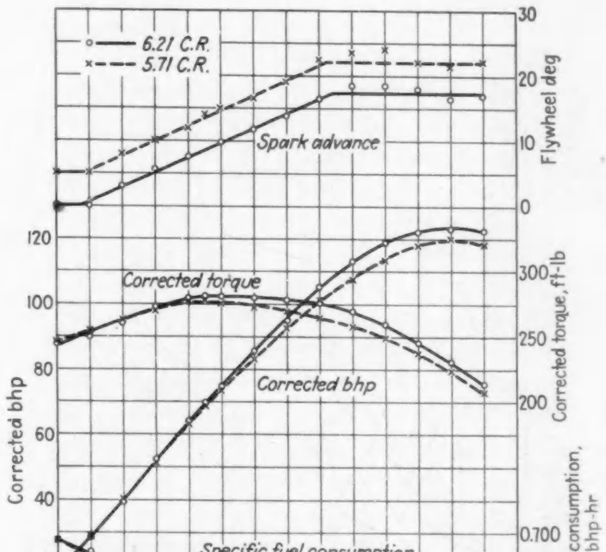
■ Fig. 6 – Full-load comparison – standard spark and fuel versus increased fuel and spark set at each speed – 6.21:1 actual compression ratio – 69.5-octane fuel

economies at a fixed throttle on a level road (burette consumption over a measured mile) and varying throttles as necessitated by the terrain over a 28-mile closed-circuit course. The fuel consumption in the latter case was measured by weight. It is to be stated here that the data checked within 300 total engine revolutions for each run. Check runs were made on each point shown that were within 10 sec of each other for the total distance. I am mentioning this point to emphasize that they are controlled tests that can be duplicated. See Figs. 1 and 2.

Table 1—Fuel Analysis

	Fuel No. 1	Fuel No. 2
Specific Gravity	64.4	63.0
ASTM Octane No.	69.5	74.5
Research Octane No.	72.0	79.5
Tetraethyl Lead Content	0.6 cc per gal	1.2 cc per gal
Vapor Pressure	7 psi	9 psi
Gum Content	2 mg	3 mg
Distillation, F		
Initial	92	93
10%	128	124
50%	216	223
90%	305	336
Final	348	415

The fuels contain ½ to 1% lubricant in gasoline.

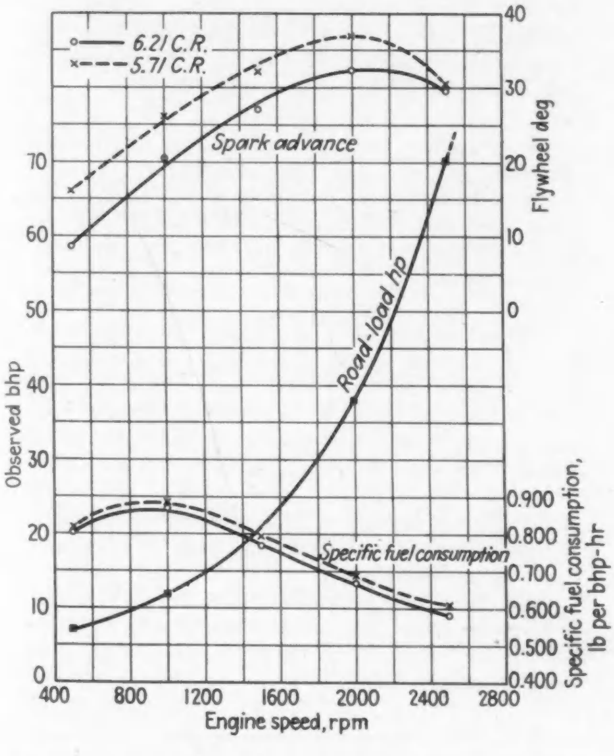


■ Fig. 7 – Full-load comparison – 6.21 (actual) versus 5.71:1 compression ratio – 69.5-octane fuel – spark set at 1300 rpm with standard distributor

Comparison tests were made on the dynamometer using the same two fuels with two compression ratios – the nominal standard 6.28:1, and an optional 5.7:1. The actual measured compression ratio of the engine used for the tests indicated in Figs. 1 and 2 is 6.25:5; for the tests of Figs. 3 to 8, 6.21:1. The standard distributor was run with the fixed setting and then the spark was set at each speed in both cases to determine if a new distributor calibration was needed. The carburetor was also recalibrated for both compression ratios in order to see if a new carburetor setting was to be desired. It should be stated that we use for road-load purposes curves based on actual vehicle requirements as determined from the road and dynamometer. Therefore, we always work to the same road-load horsepower, depending on the actual horsepower requirements of the various models. See Figs. 3 to 9 inclusive.

From a study of the foregoing results, we have drawn the following conclusions:

1. If an operator has a schedule utilizing not over 2400



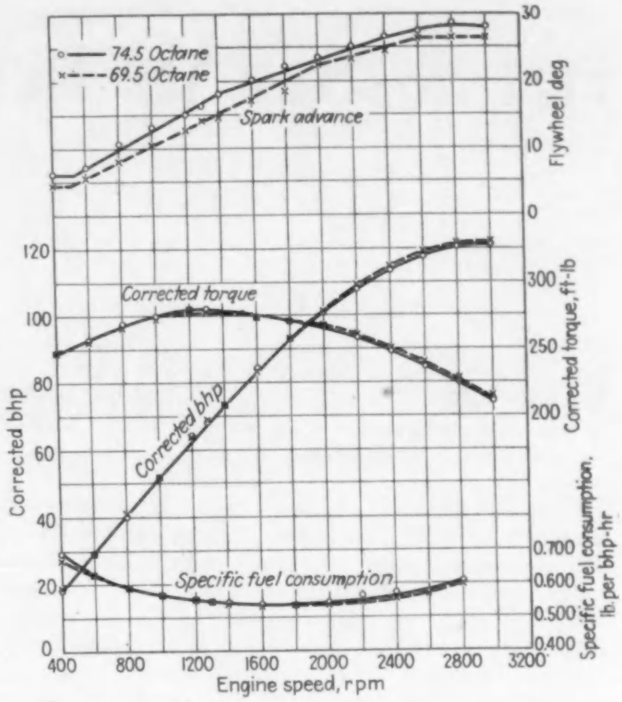
■ Fig. 8 – Road-load comparison – 6.21 (actual) versus 5.71:1 compression ratio – 69.5-octane fuel – spark set at 1300 rpm with standard distributor

rpm as a governor setting, he can simply retard his distributor, keeping the compression ratio the same, but he will lose 8% in economy by going to the lower-octane fuel.

2. If an operator has a schedule utilizing 3000 rpm as governor setting and he simply retards his spark, he will lose approximately 5% in economy.

3. With the high-speed schedule, if he wants to modify his distributor, he can obtain better top end power.

4. With either schedule, if he wants to enrich his carburetor, he can produce the same performance as with the



■ Fig. 9 – Full-load comparison – 69.5 versus 74.5-octane fuel – spark set at each speed – 5.71:1 compresion ratio

higher-octane fuel if he is willing to sacrifice 10% on economy.

5. Because of the power and economy loss as a result of changing from 6.28 to 5.71 compression ratio, we do not think the change to the lower compression ratio should be made.

6. It is to be noted that using the 74.5-octane fuel on the 5.7 compression ratio does not produce a significant increase in power and, in view of even better than 74.5-octane fuels being available after the war when quite a bit of this equipment will still be running, again it would be wrong to reduce the compression ratio.

7. On the basis of an operation of 100,000 miles per year and with gasoline at a base price of 15¢ per gal, the use of 68-70-octane fuel would cost an operator approximately $150 per year more than he is now paying for 74.5-octane fuel per unit.

It is to be emphasized that, if an operator does nothing to his units and allows violent detonation to occur, his operating costs are going to be increased by failures of head gaskets, pistons, valves, rings and bearings, and cylinder wear will be accelerated, which will still further increase both fuel and oil consumption.

PART III

of Symposium

on following pages

Part III—From the Fleet Operator's Viewpoint

by F. L. FAULKNER
Manager, Automotive Department, Armour & Co.

STRESSING the vital necessity of keeping commercial fleets operating continuously and efficiently for the duration of the war, Mr. Faulkner recommends that as much of the tetraethyl lead as will be available for essential civilian use should be set aside for a special-purpose fuel of approximately 75 octane number and made available for commercial operators who certify that 75 octane fuel is imperative to run their equipment.

Mr. Faulkner further suggests an all-purpose gasoline of not less than 65 octane, ASTM method, to be made available for civilian use in passenger cars and 1½ ton trucks which classification comprises approximately 90% of the total vehicles in use.

"When consideration is given to the fact that the majority of motor vehicles are operated at part throttle and light loads, we must recognize that a part at least of this octane hysteria has been brought about unnecessarily and has contributed to wasteful practices under a domestic economy," Mr. Faulkner avers. "The public at large will soon learn that it is unnecessary to make a race at every red light and that, if they accelerate at much lower rates, they can operate satisfactorily and smoothly with a much lower octane rated fuel than that which they have been accustomed to using, all without necessitating any material mechanical changes."

IN passing from a domestic to a war-time economy we will be confronted with many complex problems in connection with the operation of automotive vehicles. One of the most serious no doubt concerns the impending reduction in octane number of motor gasoline.

With our practically unlimited resources of petroleum refining and distribution facilities we, as operators, were little concerned as regards the availability of suitable motor gasoline until suddenly we were awakened by a limitation order restricting the sale of gasoline along the eastern seaboard. The order has been rescinded, but the effect still lingers and indicates "what can happen here." Two things may happen affecting motor gasoline, either or both of which will have a very disturbing effect on the operation of all types of motor vehicles: first, limitations on gasoline sales for restricted use and, second, further changes in motor-gasoline specifications. My comments will be limited entirely to the impending change in motor-gasoline specifications and the resultant effect on vehicle operations.

With the rapid growth of the automobile industry and increase of vehicle use has developed a constantly increased demand for improved vehicle performance and economy. There has been a remarkable response from both the vehicle manufacturers and the petroleum industry to meet and surpass these demands.

Many differences of opinion exist among the technicians in the two industries as to how improvements in economy and performance can best be obtained. However, we as operators or users are interested in this problem from the standpoint of obtaining the best overall results at a minimum of expense.

I question if operators as a class fully realize the built-in value of the products furnished both by the automotive and the petroleum industry and, further, if we are making the most advantageous use of such products. From observation it appears that a large percentage of operators are leaning somewhat heavily on these industries to carry a large share of their maintenance responsibility. Simply too much criticism is directed against the producers for operating difficulties which can be traced largely to poor maintenance practices. It is only natural to follow the lines of least resistance and, as new products are developed and offered which seemingly have the remedial benefits for certain applications, to extend the use of such products to a far greater extent than is justified.

The majority of us who are operating commercial fleets today have lived with this development throughout, and are conversant with the various ramifications of the subject. However, it will be well to review some of the earlier specifications in order to compare the possible effect that the proposed changes in motor-gasoline specifications will have on our operations.

So that we may all approach this problem from the same viewpoint, it is important to review the classification of motor vehicles in service. The 1940 registration showed 27,434,979 passenger cars, and 4,590,386 trucks and buses, totaling 32,025,365 vehicles. The 1941 estimate indicates over 34,000,000 vehicles in operation, which will use in excess of 25 billion gallons of motor gasoline annually. Forty-one per cent of these motor vehicles registered are on the farms and in towns under 2500 population, and 92% of all trucks registered were 1½ tons and under. Further, we are scrapping an average of less than 2½ million vehicles per annum, leaving us a country-wide fleet the average age of which is approximately seven years. I quote these statistics with only one thought in mind – that it is imperative in discussing this impending change in gasoline specifications to consider the groups that will be affected

Table 1—VV-M-571a—April 16, 1936—Federal Specification for Motor Fuel V

Test	Limit	Test Number F.S.B.	Test Number ASTM
Distillation	....	100.14	D86-30
Per Cent Evaporated at 65 C (149 F) (Min.)	10	.....	....
Per Cent Evaporated at 125 C (257 F) (Min.)	50	.....	....
Per Cent Evaporated at 180 C (356 F) (Min.)	90	.....	....
Residue (Max.), %	2	.....	....
Vapor Pressure at 100 F psi (Max.)	10	120.1	D323-32T
Sulfur (Max.), %	0.10	520.12	D90-30T
ASTM Octane No. (Min.)	65	600.11	D357-34T
Corrosion	None	530.23	D130-30

Note 1—Per cent evaporated shall be found by adding the distillation loss to the amount collected in the receiver at each specification temperature.

Note 2—In order to obtain comparable engine performance at different altitudes the temperatures corresponding to the various percentages evaporated shall have the maximum values as shown in the Appendix.[1]

and, for the purpose of this discussion, it is necessary to segregate the national fleet into three groups: first, all passenger cars; second, all commercial vehicles 1½ tons and under; and third, buses and trucks 2 tons and over.

The impending change in motor-gasoline specifications concerns, as I understand it, only one item of the prevailing specifications, and that is the octane rating, which is being brought about due to restrictions which the Office of Production Management has placed on raw-material supplies for the manufacture of tetraethyl lead. Current trade papers state that this curtailment has reduced current production of tetraethyl lead for domestic use to about 60% of existing demand, and still further curtailment is being discussed, which would bring the lead available for antiknock fluid down to less than half of the indicated demand. All of this leads to the conclusion that, if serious restrictions are imposed upon lead for use as an antiknock material in motor gasoline, certain groups of operators will be confronted with a very serious problem.

In our "all-out defense of the nation" it is necessary to conserve critical material of which lead is one, and it behooves us as patriotic citizens and engineers to treat this problem in a manner that will give the maximum results with a minimum waste of critical materials.

I do not feel that we of the operating group are justified in spending too much time discussing the difficulties in which group 1 (passenger cars) will find themselves due to a lowering of octane rating of their motor gasoline. All of the prevailing discussion regarding loss of efficiency (questionable to many operators) in operating passenger cars, and increased refining facilities required to make up for the wasted fuel in operating passenger cars with gasoline with a lower octane rating, will go for naught if we are confronted with use restrictions, which are likely to follow, if the rubber order is any criterion.

Group 2 (1½-ton trucks and under) are largely in the same category as Group 1. These vehicles in the main are not highly stressed and can continue to operate as in the past on a lower-octane grade of fuel.

Group 3 presents an entirely different problem, and it is in this class of vehicles that the operators are most vitally interested. Any problems which will be discussed in connection with this group are common with vehicles of Groups 1 and 2, which are operating under highly stressed conditions.

[1] Copies of this Appendix are available on request to the Society of Automotive Engineers, 29 West 39th St., New York, N. Y.

For the purpose of this discussion, I am assuming that, for the duration of the war, refiners will be limited to the production of one all-purpose domestic fuel, with an ASTM octane rating of approximately 65 and, it is hoped, a second gasoline for "special-purpose" use with an ASTM octane rating of approximately 75.

Throughout the majority of gasoline marketing areas, service stations sell three grades of gasoline, namely, 80-82 octane ethyl, 72-74 octane regular, 63-66 octane third grade, and tank-lot purchasers, in addition to the foregoing, are offered 60 octane and below, and 60-62, 400 F end point. In most marketing areas, the gravity designation is not referred to as the motoring public has been educated largely to purchase on an octane basis or, in the language of the service station, "third grade," "regular," and so on. Many commercial operators question the captioning of the 63-66 octane fuel as third grade as, in many instances, it has other qualities that may be more desirable for commercial use than have other available grades, which implies that there are factors to consider when purchasing gasoline other than octane rating alone.

Large buyers of motor gasoline usually purchase it on specification, and the very largest of these purchasers is the Federal Government, which for many years maintained gasoline specifications with no octane requirement stipulated. The first recognition of an octane rating of government fuel appeared in specification VV-M-571a, April 16, 1936, covering motor gasoline wherein a 65 minimum octane ASTM was specified. Table 1 shows the important specifications; full detail will be found in an Appendix[1]. This gasoline was termed Motor Fuel V, the specifications for which were continued in effect until Feb. 15, 1940, revisions being made under specification VV-M-564 – Feb. 15, 1940, Motor Fuel R, covering two grades of gasoline – R70 regular grade with 70 ASTM octane and R77 with 77 octane minimum. The complete specifications are otherwise similar to VV-M-571a. Also shown in the Appendix[1] under II and III are specification references for VV-M-571b, covering Motor Fuel V – 80 octane, and VV-M-564, covering Motor Fuel R72 octane, both dated March 3, 1941.

Army Specification No. 2-103, dated July 26, 1941, covering "all-purpose" motor fuel, with all principal specifications remaining the same as Motor Fuel R, with the exception of ASTM octane number which was raised to 80 minimum, was issued, as I understand it, primarily for gasoline supplied to combat units for maneuver purposes.

Table 2—Armour & Co. Specifications for Tank-Car Purchases of Gasoline—Q. C. 3025 Spec. No. G-1

Test	Standard Winter Grade	Standard Summer Grade	Source of Test	Source of Standard
I. Distillation:				
10%	120-140 F.	140-160 F.	ASTM Standard Method D-86	Armour
50%	230-250	240-260		
90%	345-365	355-375		
II. Doctor	Sweet	Sweet	U. S. Navy Dept. Spec. Method 7 Gla (7-2-23)	Armour
III. Octane No.	65 Min.	65 Min.	ASTM Tentative Method D357-34	Armour
IV. Reid Vapor Pressure, psi	12.0 (max.)	8.0 (max.)	ASTM Tentative Method D323-32T	Armour
V. Sulfur	Sl. Reaction	Sl. Reaction	ASTM Standard Method D130-30	Armour
VI. Gum mg/100 ml	5	5	ASTM Standard Method D-381-36	Armour
VII. Lead Tetraethyl, ml/gal	2.0 (max.)	2.0 (max.)	Armour	Armour

(The winter grade goes into effect on November 1 and applies until April 30. The summer grade goes into effect on May 1 and applies until October 31.)

Obviously, prior to 1940, government vehicles could be operated on approximately a 65-octane rated gasoline, and likewise, millions of other motor vehicles throughout the United States; in fact, the rural group, which represents a large block of gasoline users, have operated road vehicles as well as farm tractors satisfactorily on a gasoline rated under 65 octane.

Many operators, such as I for one, have held to a 65-octane rated gasoline. The full specifications are shown in Table 2 (IV of the Appendix). I bring this point out merely to indicate that vehicles, the average age of which is over six years, with normal duty and maintained with some degree of consistency, can be operated satisfactorily and economically on a fuel the octane rating of which is in keeping with the engine requirements.

The preponderance of equipment of large fleets, and of individual users as well, falls into the light-duty class, both in passenger cars and trucks, and there is no practical reason why, if they are in good mechanical condition, they cannot be operated both satisfactorily and economically on a 65-octane rated gasoline.

It is quite possible that the operating public has been somewhat oversold on high ASTM octane, whereas greater consideration should be given to the road octane requirements of the individual vehicle and, for your consideration, I submit the following checks that were made on 1940-model cars to determine their actual road octane requirement. All vehicles were removed from regular daily service, engines freed of carbon, but no mechanical adjustments were made:

Nine *A* cars, 1940 models, showed an average road octane requirement of 55.5.

Eleven *B* cars, 1940 models, showed an average road octane requirement of 68.4.

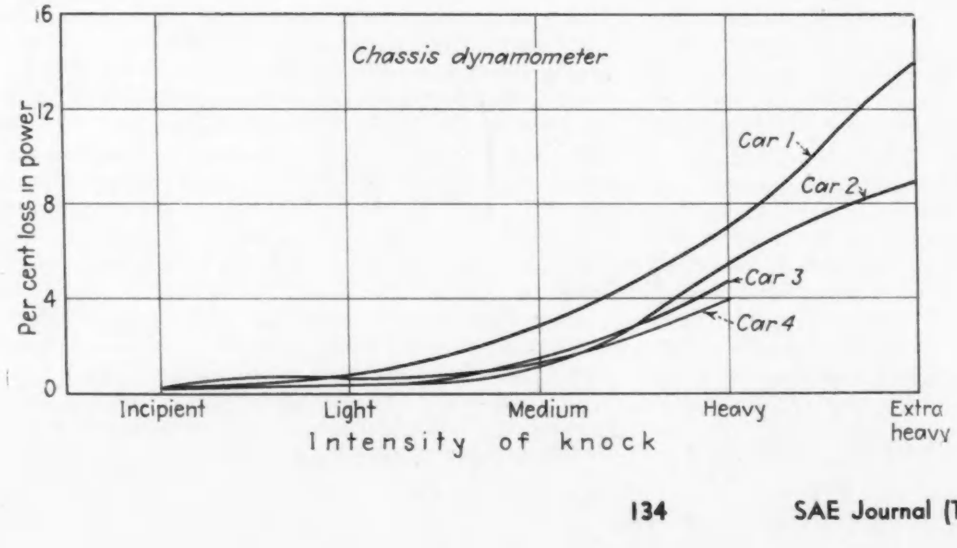
Ten *C* cars, 1940 models, showed an average road octane requirement of 63.6.

Three *D* cars, 1940 models, showed an average road octane requirement of 64.

Obviously, in the foregoing group, there would be a few cars that would be troublesome on 65-octane gasoline unless certain mechanical adjustments were made to reduce their road octane requirement. It has been general practice among operators to adjust ignition timing to suit the fuel being used. However, there has been a great deal of criticism directed toward this practice. No doubt this criticism is justified in some cases but, in general, if the engine is in good mechanical condition and the fuel that is being used has caused the engine to produce a knock of noticeable intensity, it is good practice to retune and time to a point of knock elimination.

Neil MacCoull, of The Texas Co., in his paper[2] given at the SAE Annual Meeting on Jan. 12, 1939: "Power Loss Accompanying Detonation," states: "From Fig. 2 (reproduced as Fig. 1 in this paper), it will be seen that hardly 2% in power loss occurred with a 'medium' knock as defined by the car tester, but Fig. 3 (Fig. 2, this paper) shows that this small power loss accompanied use of a gasoline of almost 15 octane numbers lower value than

[2] See SAE Transactions, April, 1939, pp. 154-160: "Power Loss Accompanying Detonation," by Neil MacCoull.

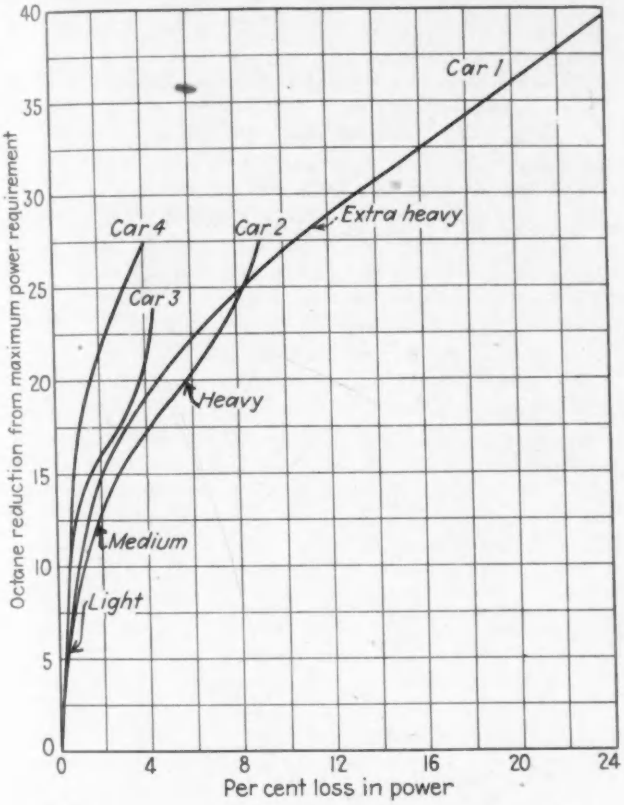


■ Fig. 1 – Effect of knock intensity on per cent loss in power at 20 mph for various cars – chassis dynamometer (Fig. 2 of "Power Loss Accompanying Detonation," by Neil MacCoull[2])

required for complete absence of knock. Such a conclusion is, of course, very general, for the reduction in octane requirement in these cars covered a range from 14 to 22 octane units for the 2% power loss mentioned, but it was sufficient to make one pause and wonder if it is economically necessary for a car to be completely free from knock at all times."

That group of operators who are concerned about loss of power while operating with a slight knock can be assured from Mr. MacCoull's data that they can operate with a considerable knock without material loss of power.

For that group of operators preferring to retard the spark in order to eliminate knock, I will refer to the paper[3] presented before the Annual Meeting of the Society on Jan. 12, 1939, by L. E. Hebl and T. B. Rendel, on "Spark

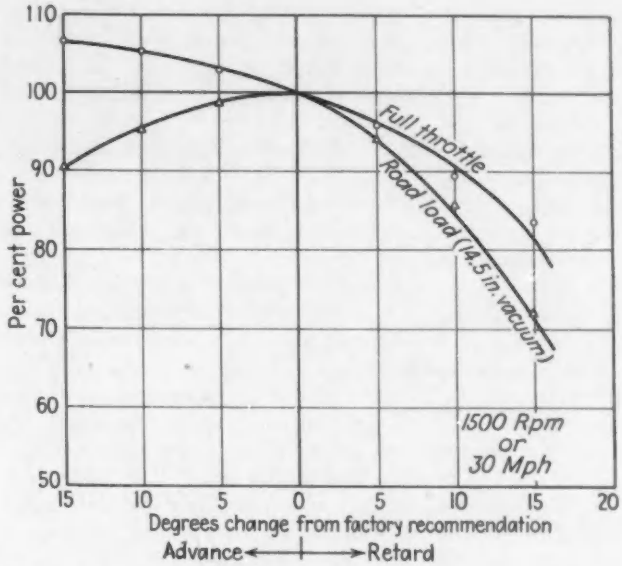


■ **Fig. 2 – Loss of engine power due to detonation from reduction in octane value at 20 mph (Fig. 3 of "Power Loss Accompanying Detonation," by Neil MacCoull[2])**

Timing – Its Relation to Road Octane Numbers and Performance," wherein they state:

"Looking back on the whole problem for a moment, it now has been shown that the wide range of present-day octane requirements is due to the flexibility in the adjustment of spark-advance mechanisms, and that reasonable adjustment of these mechanisms to take care of a fairly wide range of octane numbers has a relatively small effect on the performance of the engine."

Group 3 vehicles (buses and trucks 2 tons and above), as stated before, require and warrant special consideration. This group represents a very small percentage of total vehicles in operation in the national fleet. However, the



■ **Fig. 3 – Effect of spark-timing changes on power of Engine "B" model passenger car (From study of Chicago Calumet District Transit Co., Sept. 3, 1941)**

equipment represents, in the main, the best that the automotive industry has to offer, and the preponderance of this equipment is in the hands of outstanding operating organizations, the methods and operating practices of which are rarely equalled by operators in Groups 1 and 2.

Practically every vehicle of a given class in this group is designed for a special-purpose use, with the fullest consideration given to maintaining road loads at scheduled speeds. Operators of this type of equipment have taken the fullest advantage of every development in the automotive and the petroleum industry and, if possible, every effort should be exerted to permit them to obtain the necessary equipment and supplies to enable them to continue to operate at a high peak of efficiency.

Among Group 3 operators we find two schools of thought – the first leans toward oversized engines, normally loaded; whereas the second group uses smaller engines operating highly stressed. The first group we call the conservative type of operator, due to the long life average of his equipment, whereas the latter type operates on a basis of a more rapid turn-over in capital investment, due to the accelerated wear factor. It is felt that some of these operators no doubt can adjust their fleets to use a lower-octane fuel than they are at present using. However, fuel in itself is not the only consideration as definite power requirements must be developed in order that acceleration rates be maintained with a minimum of noise, odor, and smoke. In the case of passenger-carrying vehicles, the health, comfort, and requirements of the passengers are the first considerations.

Many engines in this type of service are designed for road octane requirements up to 75, and further, many operators in the past few years have taken advantage of the higher octane fuels available by raising their compression ratios and ignition timing above the manufacturers' recommendations for standard operations, in order to utilize the horsepower increase possible. To visualize the effect of advancing or retarding the spark on power output, data are shown in Fig. 3 which is a part of study made by the Chicago Calumet District Transit Co. in developing

[3] See SAE Transactions, May, 1939, pp. 210-218: "Spark Timing – Its Relation to Road Octane Numbers and Performance," by L. E. Hebl and T. B. Rendel.

"effects produced by spark-timing changes on power and economy." Fig. 3 shows the per cent of power increase or decrease on a typical passenger-car engine of "B" model by advancing or retarding the spark from factory recommendations. At road loads (14.5 in. hg) an advance of 5 deg ahead of factory recommendations shows a decrease in power of approximately 1½%. From the same curve, a 5-deg retard from factory recommendations would result in an approximate 5½% loss under the same road-load conditions.

At road loads, part-throttle operation, which is the range in which the majority of our commercial vehicles are operated, there is little noticeable effect on acceleration by retarding the spark timing as much as 7 deg, which is accompanied by a power loss of approximately 10%. At full throttle with a 5-deg advance there was a gain in power of 3% and with a 7-deg retard there is a loss of 6%.

Increases in power and economy could be made by raising the compression ratio within the limits of the engine design. It obviously would be impractical and unfortunate if fleets operating at peak performance were required to lower their engine octane requirements by retarding the spark setting or making other mechanical changes below manufacturers' recommendations, as this requirement would result in both loss of power and economy.

As implied earlier, there is more to gasoline specifications than the octane rating alone, and specification buyers maintain a control on many items, of which I desire to emphasize only two, namely, gravity and volatility.

It is next to impossible to purchase gasoline under present refinery marketing practices on a gravity basis; however, it has been our finding that, where we can obtain gasoline for fleet use having an API gravity of 57-58 with reasonable end point, as compared with gasoline having gravity of 60-62, there is a material gain in gasoline economy. Road tests that we have made show savings of around 4%.

Volatility is a very important characteristic of gasoline. In the past ten years we have experienced a very definite increase in volatility in all grades of gasoline. It is customary to determine the volatility by ASTM distillation, with particular emphasis placed on the 10%, 50% and 90% off point. Each refiner has his own standards of volatility for his various grades of gasoline, and there is no general standard in the industry.

Due to the terrific demand for passenger-car quality gasoline, the refiners have had little choice but to produce gasolines that would meet with popular favor. In this trend toward a flashy performing type of gasoline, commercial vehicle operators have suffered. Gasoline costs are second only to payroll in connection with the operating costs of a fleet of motor vehicles, and the time is no doubt opportune for serious consideration to be given to the production of a commercial-grade gasoline suitable for overall commercial-vehicle use, that has a maximum of economy possibilities.

I am not inferring that this problem has not been given serious consideration by the refiner, but heretofore, the preponderance of demand for other types of gasoline has worked to the disadvantage of the commercial operator.

The majority of commercial units are warmed up before going on duty; engines are operating more or less continuously throughout the day; and, in many operations, the work is heavy-duty and continuous with infrequent stops, all of which demands less volatile fuels. Further, average engine temperatures are being maintained at a higher level, and higher end-point fuels can be burned completely and satisfactorily with better economy. The less volatile fuels weigh more and have available more Btu per gallon.

Space does not permit discussion of all phases of this problem at this time but, for your further information, I have included in an Appendix[1], data accumulated through road tests and laboratory tests conducted in cooperation with the Standard Oil Co. (Ind.), covering five different gasolines, all of the same ASTM octane rating, but which vary in API gravity, 10%, 50%, and 90% off point.

The original purpose in making these tests was to determine whether we were justified in maintaining our volatility specification in view of the difficulty we were having in purchasing such gasoline in the open market. The test not only confirmed the soundness of our specification, but indicated the losses which certain types of commercial operators sustained in being required to use passenger-car type fuel for truck work.

You will note in referring to the data in Table I, VIII of the Appendix[1], the most satisfactory was Fuel D with initial of 106 F, 10%-158 F, 50%-284 F, 90%-367 F, weighing 6.233 lb per gal with 117,800 Btu per gal.

Fuel A, which differs from fuel D by having a slightly lower 50% point, shows up slightly more favorably under road-load conditions, but lower than fuel D under both part- and full-throttle dynamometer tests.

When consideration is given to the fact that the majority of motor vehicles are operated at part throttle, light loads, then we must recognize that a part at least of this octane hysteria has been brought about unnecessarily and has contributed to wasteful practices under a domestic economy. The public at large will soon learn that it is unnecessary to make a race at every red light and that, if they accelerate at much lower rates, they can operate satisfactorily and smoothly with a much lower octane rated fuel than they have been accustomed to using, without necessitating any material mechanical changes. I will go even further and recommend that some type of manifold pressure control be resorted to on vehicles of Groups 1 and 2 to prevent wide-open throttle operation. By a suitable stop the throttle can be so set that the vacuum cannot go below 2 in. hg which will be the equivalent of a reduction of approximately 6 octane numbers below the wide-open throttle octane requirement of the engine. This arrangement will not seriously affect the accelerating ability of the vehicle.

In conclusion I recommend that, in our approach to the solution of the present gasoline situation, we definitely consider the needs and requirements of automotive vehicles (passenger cars and trucks) used for commercial purposes as separate and apart from automobiles predominantly used for pleasure or convenience purposes.

Due to the necessary curtailment of raw materials for manufacture of tetraethyl lead I recommend that we insist that as much as necessary of the tetraethyl lead as will be made available for civilian use be set aside for a special-purpose fuel of approximately 75 octane number and made available for commercial operators as may be able to certify that a 75-octane fuel is imperative for the efficient and continued use of their equipment.

Further, that an all-purpose gasoline of not less than 65 ASTM octane number be made available for all other civilian use, which at the present is comprised of Groups 1 and 2 vehicles previously defined, which represents approximately 90% of the total.

DISCUSSION

Lower-Octane Fuel Symposium

—Emil P. Gohn
Atlantic Refining Co.

IN view of the fact that most of the vehicles which will be operating on these lower-octane fuels will be quite "carboned up" and operating under a high load factor, I should like to inquire what the possible effect on exhaust temperatures may be if the compression ratio of these engines is not changed and the spark retarded to permit operation on the lower-octane fuel. Reference is made herein to the larger displacement engines, for example 350 cu in. displacement and above.

Reply – Mr. Weider

OUR recent work has not shown any particular change in exhaust temperature between the lower and the higher octane fuel when operating at the same compression ratio and retarded spark setting. Nor have we encountered any preignition or exceptional valve trouble as a result of this change in operating conditions. However, it should be remembered that these tests were of comparatively short duration, and it is not possible to state definitely what the effects might be in long-term operation.

—Dr. Graham Edgar
Ethyl Gasoline Corp.

UNFORTUNATELY, the military demands for tetraethyl lead for the United States and its allies are so great that they do not leave as much of the product as we would like for civilian consumption. However, the group in Washington which has the most to do with allocation of raw materials hopes to accomplish the necessary allocations with the least possible dislocation of the vital industries. As a result, it seems likely at this time that we may expect to have sufficient tetraethyl lead for civilian consumption so that the fuel for the average motorist will not drop below 70 ASTM Motor Method octane rating although it will probably not be higher than 72 octane number. At the same time, it seems likely that a fuel of higher octane value will be available for at least those commercial vehicles which are dependent on such quality for the maintenance of their service.

—Dr. H. C. Dickinson
National Bureau of Standards

NATURALLY, all of the cars on the road do not have the same octane requirement. Therefore, it is important that adjustments be so made that individual engines, even the worst of them, can operate substantially without knock on whatever fuel is supplied. Otherwise, these engines will get worse and worse as time goes on so far as their octane requirements are concerned. We should also remember that engines which are free of carbon are not representative of the vehicles actually operating on the road. Any effort to keep all vehicles so cleaned of carbon that their octane requirements could be kept down to those of approximately clean engine operation would be extremely costly and bad because of the attendant wear and part replacement which would result.

— Cecil M. Billings
Gulf Oil Corp.

I RECENTLY had occasion to go over some rather interesting data relating to losses in efficiency which indicated that improper lubrication of the transmission and axle gears, particularly when churning of the lubricant under load occurs, may result in as much as 50% reduction in efficiency. Thus, the use of the proper lubricants in their equipment is one way in which operators may assist in getting the maximum out of their vehicles during this emergency period. I should like to ask Mr. Weider what was the air-fuel ratio used in the tests which he described, which I believe he stated were run under optimum conditions.

Reply – Mr. Weider

IT is impossible for me to give the exact air-fuel ratio because we didn't check it during these tests. However, similar work on other super-power engines has shown the optimum economy air-fuel ratio to be about 16.5:1 at part load.

—A. G. Marshall
Shell Oil Co., Inc.

MR. FAULKNER made a strong plea for heavier fuels for fleet operation in his paper. In addition to the drop in octane ratings pointed out by Mr. Hubner, the lighter fractions of present motor fuels are going to be taken out to provide increased supplies of aviation base fuel, and also the lighter fractions of present cracked stocks are going to be taken out to give additional charging material for the production of alkylate. These changes probably will mean that the fuel for civilian use will be heavier than it has been in the past so that Mr. Faulkner is going to get his wish and have ample opportunity to use less volatile fuels in fleet operation.

—Chauncey W. Smith
University of Nebraska

A REDUCTION in octane number is something which should be considered with reference to its effect on tractor operation. If fuels with the lowest octane numbers suggested, namely 68 and below, become the only available fuels for tractor use, it will hurt. On the other hand, if the more optimistic views prevail and we have available fuels with octane numbers ranging up to 72-74, then the tractor industry will have no problem in this field because the highest octane number used in our tractor testing work this year was 73. However, many tractor engineers intend to use higher-octane fuels in the future. But, for the duration of the emergency, we do not look for much development work.

—Austin M. Wolf
Automotive Consultant

IN the present situation it is particularly important that operators keep their equipment in the best possible condition so that they can conserve the Btu's in their fuel. One important thing in this conservation effort will be maintaining the best possible load factor. If necessary, some redistribution of trucks and buses through localities in accordance with the load conditions might be arranged.

—Merrill C. Horine
Mack Mfg. Corp.

WE are indebted to Mr. Weider for tackling the problem of operation on lower-octane fuels so promptly. At the same time, the entire motor industry is to be indicted for not being better prepared for this situation. One thing which war may force us to do is look into some of the things that we have tended to neglect during the era of rising compression ratios, rising octane values, and the power gains which they promised. The remarks of Mr. Billings are particularly timely. We must improve the economy of our vehicles, and more attention to friction losses is a very important part of this job. We agree with Mr. Billings' statement that a lot of this loss is due to oil churning. Several ways of correcting this condition are being studied. One suggestion is to go to dry-sump lubrication of transmissions and axles. Another point which should be carefully examined is the intake temperature. If we have heavier fuels, we must use more heat to burn them under certain conditions. But we must take care to control this increased

heat. If the present situation forces us to look into these problems, it will be some compensation for the restrictions which war is placing on us.

—H. L. Eberts

Montreal Tramways Co.

WE, in Canada, have been facing for some time these same problems with which you are just beginning to deal. There are a number of new industrial plants in Canada in areas which are not served by any railroad or bus facilities and which are entirely dependent upon motor transport. As a result, our company among others has been expanding its bus fleets greatly to take care of these plants. All of the new units which we have added are either 2-stroke diesels or high-compression gasoline engines. The latter units require a fuel of 76 to 78 octane number. We can't retard the spark so as to operate them on lower-octane gasoline because we run into excessive valve trouble during intermittent operation such as that in which these units are used. Then, too, during acceleration from idle, we get rather severe knock. Because of the demands of the new industrial plants for skilled workers, we find it increasingly difficult to obtain mechanics for service work in the maintenance of these engines. New alloy pistons, because of the war demands for these raw materials and finished products, are difficult to get. We have, therefore, taken this problem up with the Canadian Government and it has arranged to have us supplied with 68 to 70 octane fuel for our older gasoline units and 76 to 78 octane fuel for the newer units. Since the overall efficiency of the industry is the main concern of all of us, it is imperative that essential equipment be kept running with a minimum amount of maintenance. Thus, the fuel situation has many more ramifications than just how the motor sounds. It is the shortage of skilled mechanics and replacement parts that makes it now more imperative than ever before to operate our equipment on better fuels.

—E. J. Gay

Ethyl Gasoline Corp.

IN discussing this question of operation of heavy-duty equipment on lower-octane fuels, no comment has been or should be made on the amount of extra gasoline which such operation may require since this is, after all, not particularly pertinent at this time. During the past years, operators of heavy-duty vehicles, particularly buses used in city and inter-city transportation, have taken advantage of the increases in octane number to improve their performance, economy, and service. Similarly, engine manufacturers have made such changes in design as were consistent with the improved fuels available. Some weeks back, we gathered information from many representatives of this group of operators as to just what the availability of these higher octane fuels had meant to them. In general, they indicated that these better fuels had permitted them, during the past three years, to increase the number of passengers they carried by as much as 20% without economy loss. Now, if we must operate on lower-octane fuels, the reaction may become apparent in many ways:

(1) Already it is becoming noticeable that there is a shortage of buses for the transport of workers, estimated at about 5000 vehicles. The curtailment of civilian driving through the rationing of tires will add further to the burden of the bus systems. If the fuel on which these buses must operate is such that it entails a loss of approximately 8 to 10% in power, considerable additional equipment may be necessary to transport all of the extra passengers. But new equipment is also practically impossible to obtain.

(2) Then, too, the failures of cylinder-head gaskets, valves, spark plugs, pistons, and so on, resulting from operation on lower-octane fuels may be very acute and would further complicate the replacement-parts situation.

(3) Add to these two items Mr. Eberts' comments on the shortage of skilled mechanics to service these engines and the vital need for these men in our defense industries, and we begin to see how many sides there are to this problem of lowering the octane value of fuels for heavy-duty operation.

(4) So far as the heavy-duty truck industry is concerned, the greatest loss will be in man-hours of operation among the drivers. There are at present perhaps 800,000 over-the-road "line-haul" vehicles in service where maintenance of schedules is very important. Even a loss in time of 1 to 2% will amount in a year's time to a staggering increase in man-hours of labor among the drivers in a year.

In view of all of these facts, it seems that the heavy-duty operators at least should be provided with the kind of fuel they need to avoid imposing any additional strains on the material and labor situation.

—Harry O. Mathews

Public Utility Engineering and Service Corp.

I AM very much in agreement with Mr. Faulkner's recommendations for fuels during this emergency. We have fleets which comprise all three groups of vehicles mentioned by Mr. Faulkner. We have made tests similar to those conducted by Armour and have found that there is no particular gain in using higher-octane fuels in our Class 1 and Class 2 vehicles although there are definite gains to be made in using these fuels in heavy-duty operation. By increasing the compression ratio and using higher-octane fuels, we get better economy and performance in heavy-duty service. It is, therefore, most uneconomical to reduce the octane value of fuel for bus and highly stressed truck operation to anything like the 65-octane level being suggested. I would estimate that these vehicles would suffer an overall loss of 15% if forced to operate on fuels of such low quality. Their fuel requirement would also be increased enormously, which would simply add to the general burden. However, I do believe that smaller and lighter vehicles can be operated satisfactorily on fuel approximating the present third-grade type in octane value.

Welding Keeps Them Rolling

WELDING has played an important role in automotive maintenance. In normal times it is recognized as an invaluable time saver, restoring fenders, body panels, motor pans, crankcases, body cowls, and innumerable other parts of vehicles.

Today I believe that one of the first steps that can be taken by the operator to help himself in getting his vehicles back on the road is to make a study of what is going into the junk boxes. These containers may become potential stockrooms, and if he has not already done so, he should start to establish a psychology of thrift among his mechanics. Larger companies, aware of the mounting expense in operating motor vehicles and appreciating the savings in a salvage program, have been doing this for years. But today it must be intensified by a closer scrutiny of every worn or broken part to determine what can be done to save it. Dealers and service managers would do well to assist individual owners and the small fleet operators, whose business does not justify the overhead of a transportation manager, to appreciate what welding can accomplish. Perhaps the activity of long neglected service stations will be stimulated. Perhaps parts now on shelves could be exchanged for an old part which, in turn, could be welded and returned to stock.

In my organization, by the use of acetylene welding, we are successfully salvaging upper and lower crankcases, cylinder blocks, cylinder heads, valve rocker arms, water-jacket covers, upper and lower radiator domes, transmission cases, and shifter-head covers. The number of these parts still going into the scrap boxes would be astounding if figures were available for the country at large. Mufflers and parts relative to the entire exhaust system may be renewed successfully. True it is that a great many of the parts quite obviously require machine work after the welding to present smooth working surfaces and to restore manufacturer's tolerances. Going to the rear of the chassis, rear-axle differential carriers and ring gears may be saved. Well-worn brake drums can be re-used by shrinking a band around them and machining a new surface inside. Brake-rod clevises and brake arms can be built up by filling the holes and redrilling. This applies also to brake shoes and their related linkage.

Excerpts from the paper: "Welding in Maintenance," by R. H. Clark, Consolidated Edison Co., presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 13, 1942.

BUILDING UP of WORN PARTS

by W. J. CUMMING
Automotive Engineer,
Surface Transportation Corp. of New York

FACED as it is with priorities which already are affecting the procurement of replacement materials and causing a scarcity of many important metals, the next few years may well be remembered in the history of our industry as the "Years of Salvage." Under existing conditions, it is quite important to consider what salvage methods can provide the greatest benefit to automotive maintenance. The answer is probably metal coating and hard surfacing, for both methods not only allow the original dimensions to be restored economically, but furnish an added dividend, in many cases, by providing superior wearing qualities in the substitute metal.

Metal Coating

In 1923, a commercially sound process for metal coating was perfected as protection against atmospheric corrosion or chemical action. In later years, this same process was used successfully to replace mechanical wear, for increasing dimension or weight, and for altering shapes or repairing surface defects. In 1932, the writer, faced with a famine such as we now face but brought about by obsolescence rather than national defense, was forced to investigate thoroughly this excellent salvage process. It was found to possess many possibilities in the automotive field, mainly for the purpose of replacing mechanical wear.

The process was first thought impractical because there was no fusing action between the coating materials and the base material. However, it was proved later, if each problem in metal coating received careful study and certain rules were followed, excellent results might be expected. It is important in automotive practice to be positive that sprayed parts will not be stressed excessively, and that no fatigue cracks exist in the foundation piece before spraying. Further, the preparation of the foundation piece should assure keying and dovetailing deep enough to withstand the mechanical pressures involved. Also, the base material must be clean and free of oxides, oil, dirt, and water, and the sprayed metal must be finely atomized, to insure the molten particles being small enough to penetrate the finest openings in the prepared foundation and eliminate the possibility of bridging the keys.

The anchorage for metal coatings may be prepared by blasting with sand or steel grit, or by special knurling or grooving tools, depending on the later functions of the foundation piece, the metal in the base, or the material being sprayed.

The structure of sprayed metal is built up *as* small molten particles blown from a gun strike the prepared base, flatten out, and are cooled instantly. Thus, a stratified structure of small, flat, interlocking metal particles, is formed. The heat generated by the spraying operation on the outside surface of shafts or rods has a considerable effect on the anchorage between the base and the coating materials. When the first layer of sprayed metal is applied, it has a much higher temperature than the foundation. The second layer, when applied, has a higher temperature than the preceding layer; until, as each successive layer cools and contracts, the entire coating is tightened on the base piece in the same manner as a sleeve is shrunk on a shaft.

Unless special preparation and spraying technique is provided when coatings are applied to inside diameters, this same action tends to weaken the bond between base and sprayed materials. When metals are sprayed on flat surfaces, special attention must be given the preparation of the edges of the base, the thickness of each layer applied, and the selection of spraying materials with definite ductile tendencies. It may be necessary, in some cases, to spray a ductile material first, and bond the desired material to it.

In preparing the outside diameters of shafts, rods, and worn bearing areas, it is usually necessary to undercut the surface to provide a smooth area for the keying or dovetailing operations, and to insure a finished coating of uniform thickness and strength. The four steps in coating a shaft are:

1. Undercut by turning or grinding.
2. Prepare the undercut surface by blasting or machining.
3. Coat the area with the material selected.
4. Finish the coated area by turning or grinding.

There are a few simple rules in spraying technique that should be followed to assure proper results. When shafts or similar units are ready for coating, they are mounted in a lathe or other rotating device on centers, with the spray nozzle of the gun about 6 in. from the surface of the work. For base pieces 2 in. or less in diameter, the rotating or surface speed should be set at approximately 35 fpm, and the gun speed or passing speed should not be more than ⅛ in. per rotation. For larger pieces, a surface speed of 50 fpm and a carriage speed of 1/6 in. per revolution is satisfactory. If lighter coats are desired, both surface speeds and carriage speeds are increased proportionately.

Speeds for inside diameters will be about 75 fpm for surface, 1/6 in. per revolution for transverse movement.

Most flat surfaces are sprayed by hand and, while some skill is required of the operator, he soon learns to time his passes according to the materials that he is spraying. On large surfaces he will apply the materials at a rate designed to lay on a fixed amount of material per square foot.

In the maintenance field, a wide variety of worn areas can be restored successfully. The worn portion of water-pump shafts, brake camshafts, fan shafts, camshafts, crankshafts, brake hinge pins, knuckle pins, cross steering pins, compressor drive spiders and fan pulleys, generator and starter armature shafts, axle shafts, clutch shafts, and many other similar areas may be sprayed and finished in several metals. Worn bearing bores of wheels, transmissions,

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 13, 1942.]

clutches, and flywheels may be restored to original sizes. Pistons may be resized, and sleeves and bushings built up on mandrels of almost any material desired. Badly corroded water-pump housings can be covered over with non-corrosive materials without thought being given to the porous condition of the castings.

When one considers that the materials that can be sprayed include aluminum, babbitt, brass, bronze, copper, lead, monel, nickel, iron, steel (of varying hardness and both alloy and stainless), and tin or zinc, it is readily seen how an improvement may be made in the wearing quality of the materials substituted.

Another interesting feature regarding sprayed metals is the peculiar porous nature of the stratified structure, which apparently succeeds in withholding lubricants, thereby reducing wear even under heavy loads. To prove this point, Harry Shaw, before the College of Technology, Manchester, England, in 1937, said:

"Sprayed steel shafts running in white metal bearings with plain lubrication at 445 fpm speed, were tested against hardened steel shafts under the same conditions. It required four times the load to cause the sprayed shafts to seize, or a 2540-lb load as against 650 lb for the hardened steel shafts."

In analyzing spraying costs, wire size, the type of gas, and the pressures used, are factors to be taken into consideration. Generally speaking, the larger the wire used, the greater the spraying speed, and the lower the cost. The exceptions are the stiffer wires that may slow down under load, and in spraying small pieces where the edge loss may be excessive due to the greater cone of spray in larger wires.

Where they are both available, facts regarding the two gases commonly used should be known before a choice is made. Although acetylene costs more per cubic foot than does propane and a greater volume per pound of metal is used, they both require oxygen for combustion. The ratio of oxygen to acetylene for normal flame is 2.5:1, while a 6.4:1 ratio is necessary for propane, considering all metals and wire sizes. Oxygen cost may be the determining factor.

Although increased spraying speeds result from increased gas pressures, there is an economic point at which the cost of increased gas consumption exceeds the gains involved. Also, the Fire Underwriters' operating pressure limit for acetylene, in some localities, is 15 psi, and propane pressures are not restricted.

Some hint of the savings involved in coating worn areas can be gained from the following: Water-pump shafts have been sprayed with stainless steel and finished to original dimensions at one-third the cost of a new shaft, and a dividend was gained in the corrosion-resistant quality of the material substituted. Brake camshafts can be sprayed at about one-third the cost of new shafts, and the dividend is in the excellent wearing quality of the high-carbon steel coating. Six throws of an engine crankshaft can be sprayed and finished for less than one-half the cost of a new shaft, and here one finds the dividend in the long-wearing alloy-steel coating applied.

Hard Surfacing

Hard surfacing or facing is the process of welding on to wearing surfaces a coating, edge, or point of metal capable of resisting abrasion. This process can be applied equally well to new or worn parts. It is not only an excellent salvage process, but often allows weak or inadequate materials to produce greater life.

Hard-surfacing materials are usually divided into three general classes: the iron-alloy metals containing elements that increase somewhat the wear resistance and impact resistance of iron; the non-ferrous alloys that are highly abrasion-resistant; and the diamond substitutes that are essentially tungsten carbide.

The first type of materials is used generally only as filling or base materials for the non-ferrous hard-facing metals, or as a binder for the tungsten-carbide group. Since they are considerably lower in price than the other metals mentioned, it is economical to build up worn areas with this alloy, and use the non-ferrous alloy for surface hardness only. Deposits of this metal will have a tensile strength of about 40,000 psi, and the compressive strength may reach 177,000 psi. While these materials are moderately hard, they lack the excessively great wear-resisting qualities of the non-ferrous group.

The non-ferrous alloys of the second group are used almost exclusively for hard-facing against abrasion. Combined with a hardness comparable with that of hardened steel, they have excellent welding properties. It is the ability of these alloys to retain an original hardness factor, even in a "red-hot" state, that accounts for their excellent wearing quality. Tests have shown that, above 1100 F, cobalt-chromium-tungsten alloys are harder than all known alloys except the tungsten-carbide group.

There are other properties of this group that are of value in hard-facing work. The low coefficient of friction of these metals, and their tendency to take a high polish, result in less surface heat being developed in contacting parts. Thus, the life of other metals working in contact with them is increased. Wherever lubrication is difficult, the foregoing two factors are of value. Another property results in a coefficient of expansion that is similar to steel, and accounts for the unusual freedom from shrinkage cracks of applications of this group to steel.

The third class is usually considered as cutting metals; however, as inserts in varying shapes and sizes they make excellent wear-resisting surfaces if welded at critical points. They are cast in handy shapes and are held in place by a binding material, usually of the first group mentioned.

Generally, hard-facing may be accomplished by almost any standard fusing method. However, for certain applications, the oxy-acetylene method has its advantages, particularly when using the metals of the second class. When large surfaces are encountered, the metallic arc process is recommended but, in such cases, there will be a dilution with the base metal. In most cases, this dilution reduces the abrasion-resistant qualities of the hard surfacing.

The area to receive hard-facing should be free of rust, scale, and other foreign substances. These should be removed by grinding, machining, or chipping. If other cleaning methods are used, a bright surface free of oil or dirt is necessary.

When preheating is required, it should be done in a furnace or by neutral oxy-acetylene flame.

Some typical parts that have received hard-facing in automotive practice are: engine valve faces and valve seats, compressor valves and seats, clutch release yokes, universal-joint bearings, transmission shifter fingers, clutch release bearing housings, fuel-pump shafts, water-pump shafts, transmission bearing caps, valve push rods, clutch-plate pressure areas, engine rocker arm tips, camshaft cam tips, gearshift lever tips, valve tappet faces, and valve ends.

In all cases the life ratio has been increased, in some cases in a ratio as high as 10:1.

Current Problems in LIGHT AIRPLANE ENGINES

by RALPH S. WHITE

Chief, Power Plant Section, Aircraft Engineering Division, Civil Aeronautics Administration

THIS paper outlines some current problems in light airplane engines with particular reference to the 4-cyl horizontal-opposed air-cooled engines which comprise over 90% of all aircraft engines under 100 hp manufactured since 1935.

Statistics of the private flying operation from 1936-1940 are shown. Charts representing the results of the analysis and studies made by the Civil Aeronautics Administration as to the causes of all powerplant failures reported on aircraft equipped with this type of light airplane engine are given.

The method of investigating, tabulating, analyzing, and instigating corrective action of service troubles is described. The detailed handling of a persistently chronic and aggravating problem, namely, idling difficulties in flight, is discussed extensively.

The importance of the educational approach that must be applied in the handling of the manifold problems involved and as they pertain to the pilot, the airplane and airplane engine manufacturers, and the personnel of the Civil Aeronautics Administration, is discussed.

Additional service problems, such as replacement parts, icing, automobile fuel, overhaul, vibration, noise, starting, detonation, and so on, are also treated to the extent warranted by the seriousness of the problem.

In conclusion it is stated that, although the record is good, constantly improving standards of safety seem to be required to maintain this record.

THE subject of this paper is one in which I not only have a deep and abiding interest, but about which I also feel a good deal of concern. We have all heard that there is no way of judging the future except from a study of the past and, although the past record of the light airplane engine has been truly remarkable, I believe that you will agree, after taking an excursion with me to "examine the record," politically speaking, that in many respects there is much to be desired.

Before proceeding further I wish to state that any opinions expressed which have not previously been expressed by the Civil Aeronautics Administration are my own and do not necessarily express the opinion of the Civil Aeronautics Administration.

In view of the fact that light airplane engines are to be surveyed, the field for discussion was immediately narrowed to engines of low horsepower. At the time of the assignment of this paper it was also suggested that some consideration be given to the other two parts comprising the powerplant group, namely, airplane installation and propellers; accordingly, do not be surprised if I apparently make a temporary excursion off the none-too-straight nor none-too-narrow topic of engines in order to tie up the powerplant picture. It is recognized that these two subjects are topics for lengthy papers in themselves. Any attempt to define a light airplane engine would stir up too much controversy; accordingly, accompanied with a feeling of relief, the decision was made to consider the 4-cyl horizontal-opposed aircooled engines as representative of this classification inasmuch as these engines represent a type of which over 90% of all the engines under 100 hp are constructed. Since the Aircooled Motors Corp., The Aviation Corp., and the Continental Motors Corp. are the only companies actively manufacturing and selling these types of engines, all of which were rated through 1940 in the 40 to 80-hp range, their engines will be discussed collectively and will hereinafter be known in this paper as the "Little-Three" engines. It is interesting to note that the background of these three companies has a foundation of many years experience in building automotive engines. This leads me to believe that probably they approached the manifold problems involved, that is, economics, design, manufacture, quality, and so on, in essentially the same manner.

The "Little-Three" aircraft-engine manufacturers, not unlike their big-brother manufacturers who produce larger engines of the 1000-hp classification, are unhappily faced with similar problems of major magnitude if their product is to meet successfully not only current performance demands but also contemplated future requirements. The horsepower requirements alone are illustrative in this respect since, from 1936 when there were only the 40-hp engines, the horsepower delivered by this type engine has gone up approximately 20% each year.

[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., March 13, 1941.]

To support the contention that the record is truly remarkable, you are referred to Table 1 which lists the private flying operations statistics from 1936 through 1939. It is regretted that the statistics for 1940 are not included but, at the time when this paper was prepared, they were not available. These data include the operational records of all aircraft in private flying of which the aircraft equipped with the "Little-Three" engines comprise approximately two-thirds of the total number. In view of this, it is believed that the data in Table 1 are representative and give a fairly accurate picture of what the percentage breakdown would show on "Little-Three" engine equipped aircraft. Anyone who doubts that operation of airplanes equipped with the "Little-Three" engines has long since graduated from the stage of a pioneer development to an established position in routine and essential forms of transportation has only to glance over the statistical records as listed in Table 1. However, these records are not good enough and you may be assured that every feasible endeavor is being applied to assist private flying to improve steadily these impressive records. Constantly improving standards of safety are being applied as the result of the promulgation of airworthiness regulations which are considered the "yardstick" for measuring flying safety. These regulations are predicated on the basis of carefully weighing the economic factors with the factors of applying constant emphasis on safety and an insistence that no chances shall ever be taken and no precaution neglected.

Upon accepting the assignment to head up this study, I immediately asked a representative group of individuals, including the chief engineers of the "Little-Three" companies, various engineers from propeller and airplane companies, and inspection personnel who operate in the field and are daily feeling the pulse of the passing events, to contribute their beliefs and experiences as to what problems should be discussed in this paper. The immediate responses to my request, together with the overwhelming amount of information received, astonished me. They already indicate that the subject is a most important one and that it will profitably bear continued study, research, and intelligent treatment. It is hoped, in this report, to do justice to all the opinions and information submitted. A definite attempt has been made to give a generalized representative cross-section. A list of the problems that are apparently now with us was prepared and is presented as follows:

1. Failures
2. Idling
3. Icing
4. Education
5. Replacement parts
6. Fuel and oil
7. Detonation
8. Quality control
9. Overhaul
10. Starting
11. Personnel
12. Vibration
13. Noise
14. Standardization
15. Visibility
16. Welding
17. Public demand

The discussion on a few of these items such as Item 1, "failures," applies directly to the "Little-Three" engines, whereas the discussion on many of the other items, although applying directly to the "Little-Three" group may also be applied indirectly to many of the other engines. This is specifically true of Items 3, 4, 5, 8, 12, 13 and 16.

After listing all these items and, although realizing that there are many more that probably could be listed, the realization of *tempus fugit* had its effect and interrupted what otherwise would be a longer list which would probably result because of time considerations in some not even being given a veneer treatment in the general discussion. It is immediately apparent that the general numerical arrangement may be criticized, but please be reminded that the last item is not necessarily least important. However, it is believed that the first item mentioned, namely "failures," is the most important since practically all the other items are to some extent closely associated with this problem. Furthermore, it is apparent that there is a certain amount of overlapping in the listing and in the consequent treatment of these problems. Please bear with me as we "examine the record" in the numerical order as shown on the list.

1. Failures

Since the first government regulation of commercial aeronautics, reports have been received by the Civil Aeronautics Administration, formerly the Bureau of Air Commerce, and the Aeronautics Branch of the Department of

Table 1 – Private Flying Operations

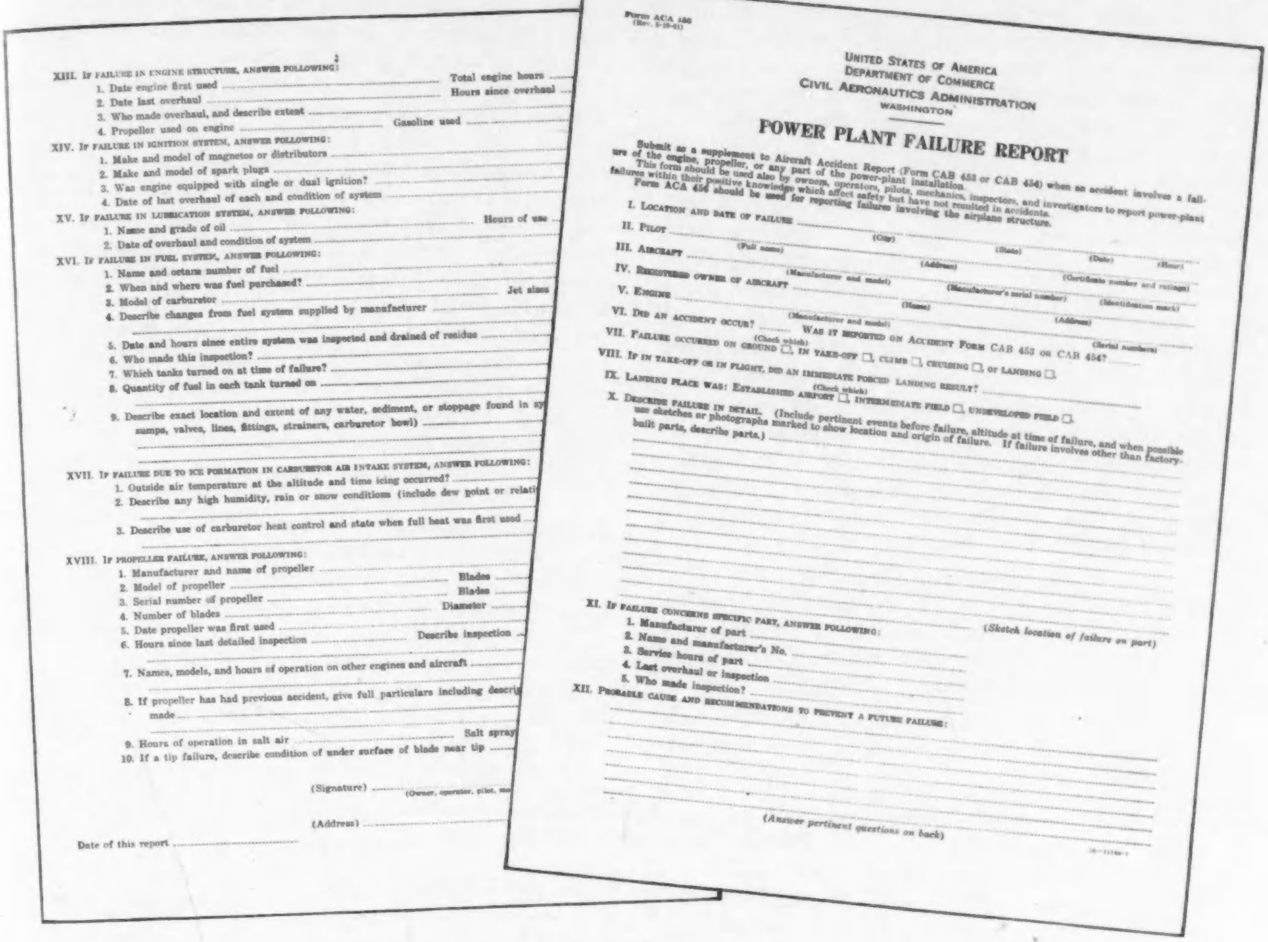
(These data were obtained from the Oct. 15, 1940, issue of the "Civil Aeronautics Journal")

	1936	1937	1938	1939	1940
Airplanes in operation (certificated and uncertificated)	8,849	10,446	10,718	12,274	—
Accidents					
Number of accidents	1,698	1,917	1,882	2,175	—
Miles flown per accident	54,959	53,832	68,735	81,778	—
Number of fatal accidents	159	185	172	194	—
Miles flown per fatal accident	586,921	557,818	752,088	916,846	—
Pilot fatalities	130	152	141	161	—
Co-pilot or student fatalities	15	16	15	7	—
Passenger fatalities	119	112	115	139	—
Aircraft crew fatalities (other than pilot, co-pilot or student)	6	2	1	4	—
Ground crew and third-party fatalities	2	1	3	3	—
Total fatalities	272	283	275	314	—
Miles flown per pilot fatality	717,849	678,923	917,440	1,104,771	—
Miles flown per passenger fatality	784,205	921,396	1,124,862	1,279,627	—
Fuel (consumed):					
Gasoline, gal.	10,451,496	10,618,240	10,201,053	16,394,335	—
Oil, gal.	316,502	310,851	287,875	460,189	—
Miles flown	93,320,375	103,196,355	129,359,095	177,868,157	—
Passengers carried:					
For hire	1,215,405	1,295,904	1,238,133	1,161,292	—
Not for hire	250,653	284,508	337,018	432,794	—
Total	1,466,058	1,580,412	1,575,151	1,594,086	—
"Little-Three" Engine Production*	800	1,020	1,670	3,640	6,000**

* This item has been added by the author to indicate the approximate number of "Little-Three" aircraft engines which have been fabricated by the Aircooled Motors Corp., The Aviation Corp., and Continental Motors Corp. The first two named companies only started to manufacture these 4-cyl horizontal-opposed engines in 1938, whereas the Continental Motors Corp. already produced this type of engine in 1931.

** This figure is estimated, since at the time this paper went to print, the complete data had not been obtained from the respective companies.

Table 2 – Form 455 for Reporting Powerplant Mechanical Trouble



Form ACA 455
(Rev. 5-10-41)

UNITED STATES OF AMERICA
DEPARTMENT OF COMMERCE
CIVIL AERONAUTICS ADMINISTRATION
WASHINGTON

POWER PLANT FAILURE REPORT

Submit as a supplement to Aircraft Accident Report (Form CAB 453 or CAB 454) when an accident involves a failure of the engine, propeller, or any part of the power-plant installation.

This form should be used also by owners, operators, pilots, mechanics, inspectors, and investigators to report power-plant failures within their positive knowledge which affect safety but have not resulted in accidents.

Form ACA 456 should be used for reporting failures involving the airplane structure.

I. LOCATION AND DATE OF FAILURE .... (City) (State) (Date) (Hour)

II. PILOT .... (Full name) (Address) (Certificate number and ratings)

III. AIRCRAFT .... (Manufacturer and model) (Manufacturer's serial number) (Identification mark)

IV. REGISTERED OWNER OF AIRCRAFT .... (Name) (Address)

V. ENGINE .... (Manufacturer and model) (Serial numbers)

VI. DID AN ACCIDENT OCCUR? .... WAS IT REPORTED ON ACCIDENT FORM CAB 453 OR CAB 454? ....

VII. FAILURE OCCURRED ON GROUND ☐, IN TAKE-OFF ☐, CLIMB ☐, CRUISING ☐, OR LANDING ☐. (Check which)

VIII. IF IN TAKE-OFF OR IN FLIGHT, DID AN IMMEDIATE FORCED LANDING RESULT? ....

IX. LANDING PLACE WAS: ESTABLISHED AIRPORT ☐, INTERMEDIATE FIELD ☐, UNDEVELOPED FIELD ☐. (Check which)

X. DESCRIBE FAILURE IN DETAIL. (Include pertinent events before failure, altitude at time of failure, and when possible use sketches or photographs marked to show location and origin of failure. If failure involves other than factory-built parts, describe parts.) ....

XI. IF FAILURE CONCERNS SPECIFIC PART, ANSWER FOLLOWING: (Sketch location of failure on part)
1. Manufacturer of part ....
2. Name and manufacturer's No. ....
3. Service hours of part ....
4. Last overhaul or inspection ....
5. Who made inspection? ....

XII. PROBABLE CAUSE AND RECOMMENDATIONS TO PREVENT A FUTURE FAILURE: ....

(Answer pertinent questions on back)

2

XIII. IF FAILURE IN ENGINE STRUCTURE, ANSWER FOLLOWING:
1. Date engine first used .... Total engine hours ....
2. Date last overhaul .... Hours since overhaul ....
3. Who made overhaul, and describe extent ....
4. Propeller used on engine .... Gasoline used ....

XIV. IF FAILURE IN IGNITION SYSTEM, ANSWER FOLLOWING:
1. Make and model of magnetos or distributors ....
2. Make and model of spark plugs ....
3. Was engine equipped with single or dual ignition? ....
4. Date of last overhaul of each and condition of system ....

XV. IF FAILURE IN LUBRICATION SYSTEM, ANSWER FOLLOWING:
1. Name and grade of oil .... Hours of use ....
2. Date of overhaul and condition of system ....

XVI. IF FAILURE IN FUEL SYSTEM, ANSWER FOLLOWING:
1. Name and octane number of fuel ....
2. When and where was fuel purchased? ....
3. Model of carburetor .... Jet sizes ....
4. Describe changes from fuel system supplied by manufacturer ....
5. Date and hours since entire system was inspected and drained of residue ....
6. Who made this inspection? ....
7. Which tanks turned on at time of failure? ....
8. Quantity of fuel in each tank turned on ....
9. Describe exact location and extent of any water, sediment, or stoppage found in sy[...] sumps, valves, lines, fittings, strainers, carburetor bowl) ....

XVII. IF FAILURE DUE TO ICE FORMATION IN CARBURETOR AIR INTAKE SYSTEM, ANSWER FOLLOWING:
1. Outside air temperature at the altitude and time icing occurred? ....
2. Describe any high humidity, rain or snow conditions (include dew point or relati[...]
3. Describe use of carburetor heat control and state when full heat was first used ....

XVIII. IF PROPELLER FAILURE, ANSWER FOLLOWING:
1. Manufacturer and name of propeller ....
2. Model of propeller .... Blades ....
3. Serial number of propeller .... Blades ....
4. Number of blades .... Diameter ....
5. Date propeller was first used ....
6. Hours since last detailed inspection .... Describe inspection ....
7. Names, models, and hours of operation on other engines and aircraft ....
8. If propeller has had previous accident, give full particulars including descri[...] made ....
9. Hours of operation in salt air .... Salt spray ....
10. If a tip failure, describe condition of under surface of blade near tip ....

(Signature) .... (Owner, operator, pilot, me[...])

(Address) ....

Date of this report ....

Commerce, concerning mechanical failures causing accidents to aircraft in service. Since 1936 the engineering analysis of mechanical failures in scheduled and miscellaneous flying has been organized so as to provide a more detailed breakdown of points of importance which can be abstracted from accidents and failures. The system now used by the Aircraft Airworthiness Section of the Administration to record the technical details of accidents and also failures gives a progressive picture of the number of failures of a particular part and the effect of the failures upon safety. This breakdown was originated first for powerplant failures and has been successful in forming a basis for corrective action and airworthiness regulations.

In this study only those failures which occurred on aircraft equipped with the "Little-Three" aircraft engines during the years 1936 to 1940 will be examined. It is well to mention at this point that two previous powerplant failure studies have been made by my predecessor, the former Chief of the Powerplant Unit, Gaylord W. Newton, now with Boeing Aircraft Co. The first paper, entitled: "A Survey of the Cause of All Powerplant Failures Reported to the Bureau of Air Commerce between January 1, 1931, and December 31, 1935," was published on June 26, 1936, as Information Paper No. 8 by the Bureau of Air Commerce. The second paper entitled: "A Survey of Mechanical Failures of Aircraft during 1936-1937," was presented by Mr. Newton at the National Aircraft Production Meeting of the Society at Los Angeles, Calif., Oct. 13 to 15, 1938. The same general treatment will be used in discussing the "failures" problem.

Because of the incompleteness of statistics concerning ground difficulties, only those failures which occurred during flight are considered. The ground troubles are an important phase of the operations and constitute a large basis for improved maintenance and safety precautions. However, the flight failures have a more immediate bearing upon safety and the reports can be considered more accurate since they are required in most instances and operators naturally make more complete investigations to determine the causes of flight difficulties.

It is believed advisable to consider first the methods by which the results of the mechanical failures in service have been received and analyzed. For miscellaneous operations Form 455 which superseded Form AC 91-5, formerly Form 202, is the medium by which reports of powerplant mechanical trouble reach the Administration. This form is required in the case of all accidents involving the powerplant, that is, where personal injury or major damage to a component of the aircraft occurs. Many other reports are received, however, of mishaps and forced landings on these forms through the field personnel of the Administration and the voluntary cooperation of the owners. A copy of Form 455 is included in Table 2. It is believed that the number of reports received is sufficiently large so that an

accurate cross-sectional picture is obtained, although perhaps only 25 to 50% of the actual troubles are reported. It should be remembered that there are about 15,000 airplanes in private and miscellaneous operation, of which approximately 10,000 comprise the "Little-Three" group. It is probably impractical for any one agency ever to obtain a complete picture of the troubles experienced in miscellaneous operation, although a well-rounded "yardstick" is obtained.

In order to understand the scope of the study, it is necessary to outline the method by which the information is taken and correlated from Form 455. Each report is studied to determine the primary failure involved and the results briefly noted on a tabulation card. A copy of this card is included as Table 3. You will note from the card that the essential particulars of the location, cause, and result of the failure can be indicated. Whereas the original incoming reports are filed under the model of airplane, engine, or propeller concerned, the tabulation cards are grouped under the subject of the failure. It is therefore possible to obtain from a single group of cards a running record of any type of failure on all aircraft arranged by date as they occur.

The tabulation card includes places for the checking of the flight condition, type of landing, and the result of failure. The following classification shows the definitions used to distinguish each item to be indicated on the card:

Mishap – is designated if there is no accident.

Accident – is designated if more than 50% of a major component of the airplane is damaged or if there are injuries incurred.

Ground – is designated if the failure occurs in the interval from leaving the blocks to starting the take-off run or while taxiing to the blocks after landing.

Take-Off – is designated if the failure occurs during the period from start of the take-off run to clearing airport obstacles or reducing to climb power.

Climb – is designated if the failure occurs when the airplane is climbing in low pitch or with more than cruising power.

Cruising – is designated if the failure occurs when the airplane is in the normal cruising attitude.

Landing – is designated if the failure occurs during the period when engine power is reduced and the airplane assumes a gliding angle for approach to landing until taxiing after landing.

Forced Landing – is designated only when the airplane is forced to an immediate landing within gliding range or a compulsory landing at the first available field with the failure preventing continuation of flight.

Precautionary Landing – is designated when the landing is other than forced or scheduled, or is a landing when the pilot has the option of other fields or continuing flight.

Scheduled Landing – is designated when the landing is made at the usual scheduled stop.

Flight Continued – is designated when the pilot continues flight to his destination (refers to private operation).

No Damage to Plane – is designated when the failure does not damage other parts of the airplane.

Airplane Damage – is designated when the failure results in damage to other parts of the airplane.

Personal Injury – tabulates the number of persons that suffered injuries.

Table 3 – Tabulation Card for Form 455

Time & Date	MISHAP ___	GROUND ___	LANDING WAS:	NO DAMAGE TO PLANE ___	ESTABLISHED AIRPORT ___	CLASSIFICATION: AIRPLANE ___ ENGINE ___ PROP. ___ EQUIPMENT ___
Place	ACCIDENT ___	TAKEOFF ___	FORCED ___	AIRPLANE DAMAGE ___	EMERGENCY FIELD ___	Airplane, name, model and serial number
Airline-Non Airline		CLIMB ___	PRECAU-TIONARY ___	PERSONAL INJURY NO. ___	UNDEVELOPED FIELD ___	Engine, prop. or equipment model
NC ___		CRUISING ___	SCHEDULED ___	FATAL INJURY NO. ___		
		LANDING ___	FLIGHT CONTINUED ___			

Description of failure

Prepared by:

Checked by:

MECHANICAL INTERRUPTION
Tabulation Card for Form AC 61-2
#2/70

Table 4 – Powerplant Failures During Miscellaneous Operation From 1936 Through 1940 Which Have Occurred on Airplanes Equipped with the "Little-Three" Airplane Engines

	1936	1937	1938	1939	1940
Engine – Structure					
Accessory Drive	—	—	—	—	2
Cam Assembly	—	—	—	—	1
Crankcase	—	—	—	2	2
Crankcase – Main Bearing	—	—	1	—	—
Crankcase Studs	—	—	—	—	1
Crankshaft	1	7	5	5	9
Cylinder Barrel	—	—	—	—	3
Cylinder Head	—	—	—	2	—
Cylinder Studs	—	—	—	—	3
Connecting Rod	—	—	—	—	4
Piston – Broken	—	—	—	—	3
Piston – Burned	—	—	—	—	5
Piston – Seized	2	6	7	10	10
Piston Pin	—	—	—	—	3
Piston – Rings Broken	—	—	—	1	—
Piston – Rings Stuck	—	—	3	2	—
Rocker Assembly	—	—	—	—	2
Valves – Exhaust Broken	—	—	—	—	8
Valves – Exhaust Stuck	—	—	—	—	2
Valves – Intake Broken	—	—	—	—	1
Valves – Intake Stuck	—	—	1	1	—
Valve Springs	—	1	—	—	—
Miscellaneous	—	—	1	5	14
Undetermined	—	—	—	—	2
Engine – Fuel System					
Idling	—	4	6	38	149 (M)
Fuel Injector	—	—	—	—	6
Carburetor	1	2	—	5	9
Miscellaneous	—	—	—	—	4
Engine – Ignition					
Magneto	5	3	7	7	20 (M)
Spark Plugs	1	2	3	9	7
Miscellaneous	—	—	—	1	1
Engine – Lubrication					
Pump	—	—	1	—	1
Temperature – High	—	—	—	1	—
Miscellaneous	—	—	—	—	1
Engine – Personnel					
Miscellaneous	1	9	9	11	17
Engine – Undetermined					
Miscellaneous	1	4	6	15	22

Note: (M) following a figure indicates that, because of chronic trouble, service bulletins or Airworthiness Maintenance Bulletins have been issued by the airplane or airplane-engine manufacturer, or by the CAA, to bring the problem under control.

Fatal Injury – tabulates the number of persons that suffered a fatal injury.

Established Airport – is designated when the landing is made at an airport established as a regular terminal for the class of ship in question.

Emergency Field – is designated when the landing is made at a Department of Commerce emergency landing field or equivalent landing field designated as such by the airlines.

Undeveloped Field – is designated when the landing is made at a location other than an established airport or emergency field.

These definitions have been found to be a convenient method of distinguishing between the various phases of flight. They also permit the important failures to be picked out at any later date by a simple reference to personnel and fatal injury and those only involving airplane damage. The classifications also correspond to the actual events which occur in airline operation and are adaptable readily to miscellaneous flying, although the distinctions in flight conditions are not as clearly defined in miscellaneous operation.

As pointed out before, the basis for the information in this survey is reports received from, and filled out by, operators and pilots. The reports are routed through various channels to the Aircraft Airworthiness Section where a detailed study is made of each report. This study includes an analysis of the report to determine the technical nature of the failure and a comparison of the new failure with any previous similar ones in the files. As a result, corrective steps are initiated when necessary and copies of the reports are forwarded to the manufacturers concerned if they have not already been advised. The filing of these reports in a single place at the Administration results in a "trouble file" being developed so that all serious difficulties experienced with aircraft, engines, or propellers in service are immediately at hand. The reports received have been the basis of new airworthiness requirements, corrective action by the Administration, and more complete inspection instructions to catch difficulties before they can occur in service.

It will possibly be of assistance to define what constitutes the powerplant installation of an airplane. The propeller and engine are obviously included, but it is more difficult to determine what associated parts should be considered as part of the installation. For the purposes of this survey all items other than the primary structure of the airplane, which are essential for the propulsion of the airplane are considered as a part of the powerplant installation. For instance, engine cowling, fuel and oil tanks, engine controls, propeller controls, and their attendant systems, are included. Some of these items are, to a certain extent, structural parts but, nevertheless, they perform their function as a supply or controlling medium for engine operation. At this time it is also advisable to point out the distinctions between failures classed as engine failures and those classed as airplane installation or propeller failures under the general heading of powerplant.

Engine failures are considered to be those which concern parts manufactured by the engine manufacturer or sup-

Table 5 – Powerplant Failures During Miscellaneous Operation From 1936 Through 1940 Which Have Occurred on Airplanes Equipped with the "Little-Three" Airplane Engines

	1936	1937	1938	1939	1940
Airplane – Accessories					
Miscellaneous	—	—	—	2	—
Airplane – Controls					
Throttle	—	1	1	1	2
Miscellaneous	—	—	—	1	6
Airplane – Cooling					
Cowling	—	4	1	—	6
Airplane – Electrical					
Miscellaneous	—	1	3	2	2
Airplane – Exhaust					
Carburetor Heater	—	—	—	—	1
Exhaust Manifold	—	—	—	1	2
Airplane – Fuel System					
Airlock in System	—	1	1	1	7
Dirt in System	7	3	6	7	15
Ice in Carburetor	1	2	4	1	39
Lines	—	—	2	2	6
Tank – Leak	—	—	—	—	1
Water In System	2	4	8	11	20
Miscellaneous	—	2	3	7	11
Undetermined	2	4	5	6	17
Airplane – Instruments					
Fuel Gage	—	—	—	—	8
Airplane – Lubrication					
High Temperature & Foaming	—	—	—	—	1
Lines	—	—	—	—	3
Tank – Leak	—	—	—	—	4
Miscellaneous	—	—	—	1	4
Airplane – Personnel					
Miscellaneous	1	1	3	20	63
Propeller – Structure					
Blade	—	2	1	1	7 (M)
Metal Hub and Bolts	—	2	5 (M)	—	2
Propeller – Personnel					
Assembly	—	1	—	4 (M)	1

Note: (M) following a figure indicates that, because of chronic trouble, service bulletins or Airworthiness Maintenance Bulletins have been issued by the airplane or airplane-engine manufacturer, or by the CAA, to bring the problem under control.

plied by him as an integral and attached part of his engine. A borderline example of a failure classed as an engine failure is that of a generator when an engine is equipped with battery ignition. If the engine had magneto ignition, this item would be classed as an accessory supplied by the aircraft manufacturer and counted as an equipment failure if an electrical failure occurred. The regulations of the Administration are separated into the divisions of engine, propeller, and the airplane portion of the powerplant installation, so this is found to be a convenient means of breaking down the survey.

Airplane failures are considered to be those which occur to portions of the powerplant installation supplied by the airplane manufacturer although, in some cases, some of the parts installed by the airplane manufacturer might be purchased from the engine manufacturer. For example, accessories added to the rear case of an engine not basically essential for the operation of the engine in flight are in the airplane portion of the powerplant installation. Also cowling, fuel-and-oil supply systems, and controls are distinctly within the province of the airplane manufacturer to design and install. They are thus considered as airplane failures and the same situation is true of propellers. It is believed that a study of the charts, Tables 4 and 5, will show the distinctions which have been made.

Tables 4 and 5 show 921 failures of a powerplant nature in the subject survey in 1936-1940, inclusive. The total is broken down into the various parts of engines which have failed as well as the airplane installation and the propeller items. It is possible to read from the charts the percentages of each major item as well as the percentage of each individual item with respect to the total. It should be mentioned at this point that additional reports were still filtering in for 1940 when this paper was prepared, but it is estimated the 1940 results will not be affected more than 5% from this cause.

The charts are broken down into as many headings as convenient in order to illustrate the types of failures considered. It was not considered advisable to break down some of the items into their component parts because of the small number of failures reported and the difficulty of showing these in their proper relationship on a large chart. Many of the items are self-explanatory but, to understand the chart more completely, it is necessary to describe some of the failures which are listed.

As may be seen in the last column of Table 6, the engine, airplane and propeller were responsible for 57, 40 and 3% of the total failures. Surprisingly these percentages did not vary much throughout the years of this survey.

Referring to the engine structural failures, we find that crankshaft difficulties are occurring but are not giving chronic trouble. It is known that a nose-over with consequent misalignment or propeller unbalance will cause a crankshaft failure with comparatively small amounts of further service. It is now possible for the manufacturers to determine resonant vibration conditions since equipment has been made available for this purpose. Engine vibration studies are now out of the realm of research and now comprise part of the development testing.

Cylinder-head failures are noticeably low. This result is due to the fact that engines in private operation have comparatively low outputs and the engines are not being subjected to continuous cruising operations which accumulate a large number of hours.

Going on to the fuel system of the engine, we find that the carburetor difficulties assumed a certain proportion which may have been due to the types of fuel and maintenance that private aircraft receive. Some cases of stuck needle valves and dirt in carburetor bowls and jets have been reported. These difficulties showed up to some extent in take-off and resulted in forced landings in almost every instance. Reference to Table 4 reveals that considerable idling difficulty occurred, so much so, in fact, that this problem is being given special treatment under Item 2, "Idling."

The magneto and spark-plug difficulties on Table 4 were largely due to the use of single ignition. Spark plugs were 1% of the total failures which is due to the fact that spark plugs in this type of operation are not operating on high-output engines, and most of the instances of spark-plug difficulties are not reported.

The lubrication difficulties are also very few due to these troubles being as a rule minor unless they happen to result in a structural failure which would be reported as an accident or a forced landing. Usually the engine will run long enough to accomplish a satisfactory landing.

The engine personnel failures were mostly cases of purely improper maintenance with some cases of pilot error in engine operation. Some of the most serious damage or injury occurred due to the pilot undershooting the field and allowing the engine to load up in the glide for landing. The measures prescribed in the idling pamphlets will prevent this result.

A comparatively large percentage of undetermined en-

Table 6 – Recapitulation of Powerplant Failures (1936-1940) on Airplanes Equipped with the "Little-Three" Airplane Engines

	1936		1937		1938		1939		1940		Total 1936-1940	
	No.	%	No.	%	No.	%	No.	%	No.	%	No.	%
Engine (Total)	12	48	38	57.5	50	54	115	60	312	57	527	57.1
Structural	3	12	14	21	18	19.5	28	14.8	75	13.7	138	15.0
Fuel System	1	4	6	9	6	6.5	43	26.6	168	31	224	24.3
Ignition	6	24	5	7.5	10	10.8	17	8.9	28	5.1	66	7.2
Lubrication	—	—	—	—	1	1.1	1	.5	2	.4	4	.4
Personnel	1	4	9	13.5	9	9.7	11	5.8	17	3.1	47	5.1
Undetermined	1	4	4	6	6	6.5	15	7.9	22	4.0	48	5.2
Airplane (Total)	13	52	23	34.5	37	40	71	37	224	41	368	40.0
Accessories	—	—	—	—	—	—	2	1	—	—	2	.2
Controls	—	—	1	1.5	1	1.1	2	1	8	1.5	12	1.3
Cooling	—	—	4	6	1	1.1	—	—	6	1.1	11	1.2
Electrical	—	—	1	1.5	3	3.2	2	1	2	.4	8	.9
Exhaust	—	—	—	—	—	—	1	.5	3	.6	4	.4
Fuel System	12	48	16	24	29	31.2	35	18.3	117	21.5	209	22.9
Instruments	—	—	—	—	—	—	—	—	8	1.5	8	.9
Lubrication	—	—	—	—	—	—	1	.5	12	2.2	13	1.4
Personnel	1	4	1	1.5	3	3.2	28	14.8	68	12.5	101	11.0
Propeller (Total)	—	0	5	7.5	6	6.5	5	2.6	10	1.8	26	2.8
Structural	—	—	4	6	6	6.5	1	.5	9	1.6	20	2.2
Personnel	—	—	1	1.5	—	—	4	2.1	1	.2	6	.6
Total	25	100	66	100	93	100	191	100	546	100	921	100

gine failures in private operation is due to the difficulty in getting reports of engine teardowns in the majority of cases. The cases are also not investigated by the operator since the private owner is merely interested in getting the engine back into service in good operating condition with the possible exception of satisfying his own curiosity to find out exactly what happened.

Predominant among all other causes of failure in private operation were the fuel-system failures of the powerplant installation in the airplane, which totaled almost 23% on Table 5. These failures clearly indicate that proper maintenance and modern design practice will prevent a large majority of these failures as the airline fuel systems show a percentage of less than one-third of this total.

Airlock difficulties were mostly due to maneuvering with low fuel supply using automobile gas, humps in the lines, or too high engine-compartment temperatures. The last cause should be given some attention in view of the close-fitting pressure type of cowling now being used in light engine installations where more thought than formerly is necessary in the design of cowl openings and baffles to ensure proper distribution of the cooling air.

Keeping dirt and water out of the fuel system is almost entirely a maintenance problem. Dirt may cause the engine to quit by clogging the lines, the strainer screen, or the carburetor jets, or it may flood the engine by getting on the needle valve or valve seat, while water entrained in the fuel may separate out at the bottom of the tank and strainer bowl to find its way ultimately to the carburetor and cause the engine to sputter and miss. The best remedy is to be careful of the fuel that you use, straining it before refueling if necessary; the provision of adequate tank sumps and line strainers; and to give careful attention to the draining of the sumps and strainers to ensure that all water and sediment are removed before each flight.

Fuel-line failures in light airplanes principally occur between the carburetor and fuel strainer. This result may be due to relative movements between these units or to improper support of the strainer, both conditions contributing to fatigue failure in the lines induced by vibration. The present trend toward the use of flexible lines between these components seems to be a correction for this difficulty.

Ice in the carburetor intake stands out predominantly when compared with the other difficulties. This condition has been corrected gradually by more emphasis on intake air-heater design in small aircraft. Since the icing problem has been and is a problem of much magnitude, it is going to be given preferential treatment as your reference to Item 3, "Icing," will indicate.

The undetermined fuel-system failures would not be so large except for the fact that, in most of the cases, no evidence remained after the accident to identify the cause of the trouble in the fuel system. The number of take-off failures and the large number of cases of airplane damage resulted in demolishing the carburetor and fuel-system parts upon impact. There were also some cases of the airplane being practically intact, but the lines were clear and no water was found in the system. This condition was probably due to the engines having drawn through sufficient fuel during the glide to clear out the difficulty, but the pilot did not know this and probably could not take advantage of it due to altitude limitations.

Among failures classed as miscellaneous and undetermined, those resulting from faulty operation of the fuel shut-off valve are conspicuous. It is important to see that the fuel shut-off valve control is adequately strong to operate the valve positively to the desired position at all times under normal service conditions. Under this group also are failures due to particles of cork from the gage float getting into the carburetor jet, primer pump leakage, and fuel flow stoppage due to clogged fuel-tank vents.

Fuel-tank gages of the rod-and-float type have caused some trouble lately either by inadvertence on the part of the pilot (who in one case didn't or couldn't see the empty mark on the rod but concluded, to his dismay, that there was plenty of gas in the tank because the rod end projected some distance above the tank) or by actual malfunctioning of the gage due to the rod being "kinked" and consequently sticking. When gages of this type are used, it is advisable so to design the gage that the bent-over portion of the rod outside the tank will rest on the filler cap or tank shell when the fuel tank is empty or, preferably, when 15-min fuel supply still remains in the tank.

With regard to the lubrication system, the employment of pressure-type cowling in current light airplane designs has resulted in many instances in higher oil temperatures being developed in service and has forced the adoption of a small oil cooler in the circuit to remedy that condition. Oil filters also are coming into use on light airplanes, especially those operating in dirty or dusty terrain, as a means to remove dirt and sludge from the oil, reduce foaming, and enhance the service life of the engine.

Oil line failure was mostly confined to the small oil pressure gage lines and resulted from fatigue or high oil pressure in cold-weather operations.

Tank leaks have occurred mostly at or near the welded junction of the filler neck to the tank body.

The oil-system failures classified as miscellaneous were mostly due to the crankcase breather line freezing over during cold weather. This condition can result in serious damage to the engine, and due precautions should be observed to preclude its occurrence. The practice of running the breather line from the crankcase into the carburetor cold-air intake to secure a clean installation was responsible last winter for many forced landings, the cold air in the intake freezing the condensate in the breather line to a solid plug of ice resulting in high crankcase pressures which, in one case, blew off the oil tank filler cap with force enough to dent the cowling; in another case, it forced the oil out through the front main bearing; and, in a third case, it resulted in an explosion. As a result of such accidents the practice of routing the crankcase breather line to the carburetor cold-air intake has been discouraged and proper precautions are now observed in routing the breather line so that it will not be directly exposed to a cold-air blast or, if so, that it is suitably lagged, and in locating the breather line discharge point down near the bottom of the cowl where the discharge is carried through the cowl "gill" opening to the outside by the engine cooling air.

Reference to Table 5 reveals that there have been 26 propeller failures throughout the 5-yr period. The fact that this breaks down on a percentage basis to less than 3% of the total powerplant failures during this period is truly remarkable and probably is due to the fact that propellers are exposed and may be inspected readily at all times. The failed propellers involved are either of the fixed-pitch wood type or adjustable type with wood blades. In 1938, some trouble was experienced with fixed-pitch wood-propeller hub-flange cracking. This trouble occurred only on those hubs which had been modified to incorporate lightener holes. This difficulty was corrected quickly by

the issuance of a service bulletin by the affected engine manufacturer. In 1939, trouble was experienced as the result of improper tension (poor maintenance) being applied to the hub bolts. This trouble also was corrected by the distribution of an Airworthiness Maintenance Bulletin supplemented by the pertinent manufacturer's service bulletin. In 1940 some blade failures were experienced as the result of cracking of the wood that originated at the inboard end of the metal tipping. In this case too, an Airworthiness Maintenance Bulletin has been distributed and it is expected that this will control effectively this type of failure.

2. Idling

Reference to Table 4 reveals that there have been 38 and 149 cases of idling difficulties in flight in 1939 and 1940 respectively. The aftermath of many conferences and much correspondence between the Civil Aeronautics Administration and the affected airplane, aircraft engine, and carburetor manufacturers culminated in the promulgation and eventual distribution of Airworthiness Maintenance Bulletin No. 41 and Certificate and Inspection Division Release No. 36 to all owners of the affected aircraft. This distribution probably represents the largest mass distribution of corrective action ever undertaken by the Administration, since probably over 8000 owners will be affected. Parts of these two pamphlets are included in this paper under Item 2. They will also be referred to in Item 4, "Education" because of their educational value. These pamphlets not only contain much information of value to operators of all automotive and aircraft equipment, but also show the extent to which it is necessary to handle a particularly aggravating problem. These pamphlets are now being printed and it is my prediction that this type of trouble will be brought under control during the latter part of 1941 as a result of the dissemination of this type of literature.

The attack upon this problem as shown by the proposed bulletin and release is to require several changes of proved value and to advise the operators of the best maintenance and operating practices which are applicable. As mentioned, this maintenance bulletin, together with a copy of the release, will be mailed to each owner of an aircraft equipped with 40- to 80-hp engines of the 4-cyl horizontal-opposed type. The release will also receive wide distribution through the mailing lists.

Although the proposed action should improve the situation effectively, it is not thought that it will prove a permanent measure in view of the extraordinary maintenance and operating procedures necessary. As indicated in the first part of the release, the affected manufacturers are expected to develop design changes which will solve this problem permanently. Such changes are particularly necessary to lessen the adverse effects produced by spins and stalls.

Some of the considerations which are now being studied by the manufacturers affected are the following, some of which, it is thought, should prove beneficial in aiding to preclude the subject type of difficulty:

a. Heavier propeller with more polar moment of inertia to produce more flywheel effect.

b. Accelerating pump in the carburetor.

c. Interconnecting the heat control with the throttle so that it will turn on the heat whenever the throttle is closed but leave the heat available for use with full throttle if desired.

d. Improved idling jet arrangement.

e. Improved intake manifold design to produce more even idling. The possibility of completely insulating the intake system to prevent temperature changes should be considered.

f. Starters usable in flight.

g. Factory-made equipment for use in cold-weather operation such as cowl closures, lagging for intake pipes, and covering for oil tanks.

h. Aerodynamic changes in the airplane so that suitable gliding angles for landing approaches may be maintained without so much dependence upon the drag of an idling propeller. This arrangement would permit the use of higher idling speeds.

i. Interconnection of throttle and mixture control in this type of airplane.

Airworthiness Maintenance Bulletin No. 41 states in part:

> As a precautionary safeguard and because of service difficulties and a number of accidents resulting from improper idling operation of the engines in flight, the Civil Aeronautics Administration requests that the following changes be made on your airplane:
>
> 1. *Idling Speed* – The ground idling speed of your engine should be set to not less than 550 rpm. If the engine is rated in excess of 2700 rpm, the idling speed should be set proportionately higher. However, ground idling speeds should always be less than 700 rpm. When the setting is made, the engine should be thoroughly warm with the cockpit mixture control, if installed, in the full-rich position. The carburetor heat control should be in the "off" position when the setting is made in summer, and "on" when the setting is made in winter.
>
> 2. *Carburetor Idling System* – The idling operation of your engine should be critically observed to determine whether the carburetor idling passages may be clogged with dirt or bugs. This trouble can be detected on the ground as the engine will either fail to idle or idle improperly. If any trouble is encountered, the idling passages should be cleaned. In no case is it safe to increase the idling speed of the engine to compensate for this trouble.
>
> 3. *Carburetor Throttle Shaft Bushing* – Check your throttle shaft bushing for wear. If any appreciable looseness or shake is felt, the bushings should be replaced. Excessive clearance will result in a leak which upsets the idling operation.
>
> 4. *Carburetor Float* – Any indication of engine roughness in the idling range or fuel leakage from the carburetor may mean that your carburetor float and needle valve assembly are not functioning properly. Inspect your installation for this trouble and have your carburetor overhauled if any difficulty is apparent.
>
> 5. *Ignition System* – Burned or sticking ignition points or spark plugs in bad condition with improper gap settings may be the cause of irregular engine idling. Check the ignition points and spark plugs if any fault is suspected with them. Use only the spark plug models and gap settings recommended for your engine by the engine manufacturer.
>
> 6. *Winter Operation* – To maintain suitable oil temperatures and satisfactory idling characteristics in cold weather, it is necessary that the front opening in the cowl of your airplane for crankcase cooling be closed temporarily. In most cases the manufacturer has available suitable plates for this purpose. It is also permissible to cover the opening with doped fabric. This change should be made as soon as necessary and the opening kept closed until operation in warmer air temperatures is contemplated.
>
> Procedures outlined in this maintenance bulletin should be completed as soon as possible and a suitable entry made in your airplane log book. This entry should outline the extent of the changes made to the airplane. The entry is necessary for the information of the Civil Aeronautics Inspector.
>
> It is requested that you read the attached Certificate and Inspection Division Release No. 36 which covers the general corrective measures applicable to reducing idling difficulty in flight.
>
> It should be borne in mind that the requests and suggestions contained in this bulletin are based on the service experience of several thousand airplanes and are made in an endeavor to assist you in maintaining the original airworthiness of your airplane.
>
> In case you have sold your airplane, please forward this bulletin to the new owner.

Certificate and Inspection Division Release No. 36 entitled "Improvement of Engine Idling Characteristics in Flight" states in part:

> During the past year the Administration has received over *130 reports of engine stoppage while idling in flight*, involving all types of

aircraft equipped with 4-cyl horizontal-opposed engines of from 40 to 80 hp. In most of these reports the pilots have stated that it was not possible to re-start the engine in flight. Forced landings have therefore resulted and personal injuries have occurred in some of the cases. Undoubtedly the reports received do not represent all of the instances of this type of trouble. It is also known that there have been other cases of engines stopping which have not been reported since the engines turned freely enough to start themselves again in flight.

The Civil Aeronautics Administration is taking steps with the manufacturers concerned to obtain design improvements in new aircraft engines and their installations which will tend to eliminate these difficulties. Some of these changes undoubtedly can be applied to the powerplant installations of aircraft already in use.

It is well known that the amount an engine cools off or loads up in flight depends a good deal upon pilot operating technique. The pilots have indicated in some of the reports received that the engine was neglected for long periods of time. It is therefore logical to expect that the recent increase in flying activity has caused a corresponding increase in experience of this kind. However, it has been found that idling difficulty occurs more frequently *under operating conditions which can be controlled and improved.*

Certain measures have been found helpful by the Civil Aeronautics Administration in correcting unsatisfactory idling conditions. These measures are based upon an extensive investigation of the reports received, and include the recommendations of the aircraft, engine, and carburetor manufacturers involved. A copy of this release has been distributed with an Airworthiness Maintenance Bulletin to all owners of aircraft of from 40 to 80 hp equipped with 4-cyl horizontal-opposed engines.

Slow idling is dangerous

An idling adjustment resulting in an even slow speed on the ground is not a sufficient precaution to prevent difficulty in flight. Carburetor operation is affected by cold air, humidity, rain, angle of flights, bumps, and maneuvers. It is therefore necessary that the idling speed be sufficiently fast to allow for these variations. *Ground idling speeds for engines of this type should not be set to less than 550 rpm.* Engines with ratings in excess of 2700 rpm should be set proportionately higher. The carburetor heat should be "off" when setting idling speeds in summer and "on" when setting them in winter. The engine should be thoroughly warm with the cockpit mixture control, if installed, in the full-rich position. Excessive idling speeds should be avoided since the gliding angle is flattened to a point where spot landings become difficult.

Check idling speeds frequently

The idling speed usually tends to reduce due to wear in the throttle setscrew adjustment and operating time of the engine.

In setting the idling speed of an engine of this type, set the throttle stop screw for the desired speed and move the idling mixture adjustment until even operation and the fastest speed is obtained. Next slightly enrich the idling mixture adjustment and reset the throttle stop screw for the desired speed. Then move the throttle open and closed several times to determine that no change in the idling speed results.

Gun your engine during glides

An engine appreciates attention during a glide. Wherever possible, except during final approaches to landings and when practicing landings, it is advisable to operate at part throttle, instead of at closed throttle, in order to keep an engine warm and clear. Very little power is produced by part-throttle positions since it takes at least one-half of the rated rpm to produce one-eighth of the rated power. To keep an engine ready for use the throttle should be kept closed only for short intervals during flight. For example, after each maneuver such as a stall or a spin the engine should be thoroughly warmed up and cleared out. *In a prolonged glide the throttle should be gradually opened a sufficient amount to clear the engine at least every 250 ft of altitude lost, or about every 20 sec.* In gliding for a landing it is also advisable to clear the engine when crossing the boundary of the field so as to make sure that it will be available for use at the time of landing. To clear an engine it is necessary only to open the throttle to between 1/3 and 1/2 open position. It need only be left there for a short period to clear the spark plugs and develop heat in the intake system.

A rapid opening of the throttle should be avoided with these engines since the carburetors are not equipped with an accelerating pump. The rapid opening of the throttle produces a momentary lean mixture which may cause engine stoppage if the engine is cool. Pilots should make it a habit to ease the throttle forward until the engine picks up speed before pushing it wide open.

Your carburetor likes heat

Make it a rule to turn on your carburetor heat before closing your throttle. An engine needs heat during any extended idling operation during glides. The carburetor heater control should therefore be moved to the "full-on" position prior to closing the throttle for a landing or prior to a closed-throttle maneuver in flight in any outside air temperature. To avoid icing and idling difficulties *use full heat all of the time when the ground temperature is 50 F or below.* If desired, it is satisfactory to disconnect or secure the heat control and operate the airplane with the heat in the "full-on" position if there is no likelihood of the airplane's encountering higher temperatures during the winter months. This applies particularly to tandem airplanes where the heat control is not readily accessible from the rear seat. *Full heat should also be used when high humidity or rain is present in outside air temperatures of 70 F or less. Indications of high humidity are visible moisture, fog, cloud, or damp air. Loss in rpm is usually the first indication of icing.* If loss of rpm is experienced from this cause, apply full heat immediately. Part-open positions of the heater control are of little value in installations on aircraft of 40 to 80 hp since it is impossible to judge whether an adequate amount of heat is turned on. In most cases, little heat is added to the intake system until the valve is almost in the full-hot position.

There is a hazard involved in the use of intake heat, however, which *should not be disregarded.* In view of the adverse effect upon the performance of the airplane, heat should be used with caution during take-offs in outside temperatures above 50 F. To get ample performance from your airplane for take-off, it may be necessary to take off in dry air with the heat control "full cold" when the ground temperature is less than 50 F. This applies particularly to the 40 and 50-hp installations. When adequate take-off space is available, full heat can be used, when it is needed, in air temperatures up to 70 F. *Full heat should not be used for take-off in temperatures above 70 F unless the operating procedure for the particular model airplane requires it.*

Keep your carburetor idling system clear

A number of reports have been received concerning *erratic idling caused by the air bleed to the idling jet in the carburetor being clogged with dirt or bugs.* This trouble is noticeable on the ground since the engine will either fail to idle or idle unevenly. To correct this condition, the idling passages should be cleaned. In no case is it safe to increase the idling speed of the engine to compensate for this difficulty. Steps have been taken with the carburetor manufacturers to obtain a permanent solution to this difficulty. The Marvel carburetor is adaptable to the use of an idling air-bleed screen. It is believed that less frequent inspections of the idling system will be required when such a screen is installed. Since trouble with air bleed clogging depends upon the nature of the landing surfaces used, it is important that owners of aircraft equipped with Marvel carburetors who need this protection obtain the screen from the Marvel-Schebler Carburetor Division, Flint, Mich., or their authorized service stations.

Other sources of idling trouble

Any fuel or air leak into the carburetor or induction system will cause idling difficulty. To prevent this, engine primers should be shut off and be leak-tight. At all times in flight the primer should be locked in the "off" or the "in" position. The intake pipes and intake-pipe connections should be carefully examined to determine that the clamps are tight and no sources of leakage are present. Carburetor flange attachment bolts should be kept tight to avoid leakage between the carburetor and its mounting flange. Gaskets at this point should be renewed each time the carburetor is detached. The gasket when installed should not extend beyond the edge of the carburetor barrel.

An air leak through the throttle shaft bushings can cause erratic idling. This results from wear conditions and is normally corrected at the time of the engine overhaul. Check your throttle shaft bushings for wear. If any appreciable looseness or shake is felt, the bushings should be replaced.

It takes only a small quantity of water to stop an idling engine

Water present in fuel which has not affected the normal operation of an engine may still stop the engine when the throttle is closed. *Be water-conscious.* Check your fuel-system strainer daily for the presence of water and watch the source of the fuel put in your tanks.

Put your cockpit mixture control in the full-rich position before closing the throttle. Engines of the type under consideration do not require any use of the cockpit mixture control under 5000-ft altitude. The control, if your airplane is so equipped, should therefore be left full-rich except when flying above this altitude. Make it a habit to check the position of the mixture control and push it to the full-rich position before you close the throttle in flight.

Thoroughly warm up your engine before take-off. A cold engine may operate at take-off power but is liable to stop when the throttle is closed.

A stuck or leaking carburetor float may cause idling failures

Any indication of engine roughness in the idling range or a leaking carburetor may mean that your carburetor float is not functioning properly.

Exhaust-manifold leaks affect idling

Uneven idling may be produced by loose connections in the exhaust system.

Fuels and oils affect idling

Poor idling operation often results from the use of fuels or oils not recommended by the engine manufacturer. *Use only aviation gasoline.* No aircraft engines are at present approved for use with automobile gasoline.

Ignition

Idling difficulties are aggravated by ignition trouble. Bad or dirty spark plugs, burned or sticking ignition points, and old ignition wire cause difficulty. *Only use spark-plug models and gap settings recommended for each engine model by the engine manufacturer.* When ignition trouble is suspected, the magneto should also be checked for spark intensity.

Winter operation

In preparing an airplane of the 40 to 80-hp type for winter operation, better idling characteristics and higher oil temperatures are obtainable if the front opening in the cowling for crankcase cooling is closed temporarily during cold-weather operation. In most cases the manufacturer has available suitable plates for this purpose. It is also permissible to cover the opening with doped fabric. The opening should be uncovered when operation in warm air temperatures is contemplated. In extremely cold weather it may be necessary to lag the intake manifolds and oil tank.

Summary

Idling speed setting

1. Set not less than 550 rpm. (Avoid settings as high as 700 rpm.)
2. Check idling speed frequently for any change.

Gunning engine

1. Avoid prolonged closed-throttle operation.
2. Gun engine every 250 ft altitude or every 20 sec (in glides).
3. Open throttle gradually.

Use of carburetor heat

1. Use full heat for 50 F or below.
2. Use full heat for 70 F or below if high humidity or rain present.
3. Heat affects performance, so use it cautiously.
4. Heat not required above 70 F, except with certain airplanes.

Carburetor idling system

1. Erratic idling may mean clogged idling system.
2. Clean idling passages if engine fails to idle or idles unevenly.

Ignition system

1. Irregular idling may be caused by a faulty ignition system.
2. Check ignition points for burning or sticking.
3. Use only recommended spark plugs with proper gap settings.

Improper idling in flight can be caused by

1. Engine cooling off.
2. Slow idling.
3. Loading-up in glide.
4. Burned or sticking ignition points.
5. Lack of heat in the carburetor intake system.
6. High humidity or rain.
7. Air bleed closed in idling jet.
8. Fuel or air leak in carburetor induction system.
9. Water or sediment in fuel.
10. Mixture control not full-rich.
11. Improper warm-up.
12. Stuck float or leaking carburetor.
13. Exhaust manifold leaks.
14. Improper fuels or oils.
15. Improper spark plugs and gap settings, and leaking ignition harness.
16. Lack of winter protection measures such as closing cowling openings.

Warning – An engine which is idling improperly from any of the above causes is liable to stop in throttled flight maneuvers or when the throttle is opened too suddenly.

3. Icing

The prevention of carburetor icing is a problem that is now being attacked vigorously by both the engine and carburetor manufacturers, as well as by the airplane manufacturers. In general, the solution of this problem is being effected by the provision of an adequate amount of heat to the carburetor air, but attention is also being focused on the design details of the heater and the proper use of carburetor heat. It is gratifying to observe that the carburetor manufacturers especially are giving much more of their attention than formerly to air-intake design, realizing that this adjunct to the carburetor has a very important influence on carburetor performance, and it is hoped that this realization will result in much improved carburetor air-intake design in the near future.

In the meantime, on existing installations, attention can be directed profitably to the importance of good fits being maintained on the carburetor heater valve to exclude cold air from the induction system when full carburetor heat is applied and thus assure the carburetor of receiving the full amount of heat available.

General provisions for the prevention of carburetor icing and thus affecting intake design are outlined in Certificate & Inspection Division Release No. 21 which states in part:

General provisions

The intake and carburetor passages should be arranged, in so far as practical, consistent with design limitations, so as to avoid the formation of ice. A hot-air supply should be provided which is sufficient to permit safe operation under icing conditions, except in the case of diesel or fuel-injection engines to which special rulings are applied. All hot-air heaters should be suitably cooled when they are not in use. The intake provided for the hot air should be sufficiently sheltered to avoid clogging with snow or ice particles. The use of any screen in the hot air system is not recommended and will only be permitted where over 100 F heat rise is available and the screen is of a type which service experience has proved satisfactory. A screen may be used in the cold air intake, provided that it can be shown that no hazard exists if the screen should become clogged. . . . The hot air valve and controls should be of rugged construction and sufficiently strong to withstand the full loads which can be applied by the pilot to free the valve from an icing condition.

The amount of heat available will be determined by measuring the intake heat rise by temperature measurements of the air before it enters the carburetor. This rise is the temperature difference between the outside air and the full-hot position. The temperature rise should be determined for the condition of 30 F outside air with the airplane in a cruising attitude at 75% maximum, except take-off, power. . . .

During tests for temperature rise, care should be taken to insure that the thermometer location is such that it records the average temperature of the air flow through the intake. The results can only be relied upon when the air control valve is in the full-hot or full-cold positions due to the lack of sufficiently long passages after the valve to accomplish uniform mixing. It is also difficult to obtain an accurate temperature measurement of the hot air unless the supply is from a closed passage which has ample length to uniformly heat the air. Hot air with a uniform degree of heat is usually obtainable from a muff extending over a large portion of the exhaust system or from intensifier tubes. If the design of the hot-air intake is such that cold and hot temperature stratification takes place, measurements should be taken at several points simultaneously in the same cross-section of the air stream to obtain an average temperature. Temperature measurements taken after the venturi to substantiate adequate heat being applied ahead of the venturi to prevent icing are not favored. Such measurements in test experience have been found to vary with other conditions remaining constant and are therefore in many cases unreliable.

The use of sheltered hot air intakes may involve the loss of some intake ram and the consequent loss of power. This loss, if any, will be determined by flying the airplane in a steady climb, at best one engine inoperative rate of climb speed at maximum, except take-off, power on cold air. Then the hot air control will be quickly placed in the full-hot position and, without changing the throttle setting, the change in manifold pressure will be recorded. An equivalent test will be conducted with single-engine airplanes. If no manifold pressure gage is installed, the drop in rpm will be recorded.

Sea-level engines with conventional venturi carburetors (for example, sea-level engines with a Stromberg or Marvel circular venturi carburetor).

A minimum temperature rise of 90 F should be attainable unless the intake manifold has design features near the venturi which contribute appreciably to raising the mixture temperature or the temperature of intake passage. Where these features are present a 60 F rise may be considered satisfactory if based upon service experience under severe icing conditions. . . .

4. Education

Although the title "Education" is nice sounding, nevertheless "lack of education" (understanding) and not "education" is the problem at hand. By the term "education," I have in mind the means that should be adopted to effect the utmost in cooperation between airplane, engine, and propeller manufacturers to the end that an efficient, serviceable and airworthy powerplant installation will result; means to indoctrinate owners, operators and mechanics with knowledge of the best servicing, operating and maintenance practices which have been developed by these manufacturers; and means to imbue pilots and mechanics with the responsibility which is theirs to see that these practices are carried out in the interest of safety and serviceability. The problem is emphasized by the continual growth of the industry, although it may be said that this problem is being handled in more vigorous manner daily. There is much to be said about certain items, that is: (a) cooperation between aircraft and engine manufacturers; (b) instruction books; (c) service bulletins and Aircraft Airworthiness Bulletins; (d) general public; et al.

a. The haywire, friction-tape, and shellac type of airplane installation has eliminated itself from the present-day picture as a result of the economical factor. Utmost cooperation between the aircraft, engine, and propeller manufacturers must be maintained if the objective is a successful powerplant installation of the modern aircraft. Since the accessories used are appurtenances of the airplane and engine, it is necessary that the same degree of cooperation be practiced by the affected parties. It is necessary that the installation afford the ultimate operator a maximum overall efficiency from the composite engine-propeller-airplane combination, or else have sales affected some short time later. This is the result of the compensating features of the modern system which must also contend with the "grapevine" reports which spread very rapidly, especially when some product compares unfavorably with some competitive product. Consideration must be given to the service flight and comfort demands carefully weighed in with the safety, economic, and reliability factors. The installation of an engine in an airplane is still far from an exact science. Consequently, it is necessary that the aircraft and engine manufacturers work together very harmoniously in every way to secure proper balance in the design of installations. It is known that all successful engine manufacturers have available experienced field engineering personnel and equipment to assist the airplane manufacturer relative to the cooling, vibration, local airflow, lubrication, and all other related problems that must be solved successfully before the installation may be considered satisfactory. Apropos of this subject the following thought is considered pertinent:

> The cooling of an aircooled engine under test conditions is a function of the air flow and engine cowling design. When the engine is fitted with intercylinder baffles that control the air flow around the engine, the pressure drop across these baffles is a good index of the air flow. Aside from the design of the engine and intercylinder baffles, this air pressure drop depends upon the airplane speed, the propeller used, and shape and size of the engine cowling entrance and exits. In general, if the air flow past the cylinders is reduced, an increase in airplane performance will result. If cooling can be obtained by properly directing the flow of a small amount of air past the engine, such a method is desirable. However, this air flow should not be reduced to a point where insufficient engine cooling is obtained.

If this condition does arise in actual practice, this is where the engine-manufacturer's installation representative should start talking "engineering facts" to the airplane-manufacturer's engineers.

b. Manufacturers are realizing more and more the value of suitable instruction books to cover all contingencies that may arise. At one time it almost took an Act of Congress for an individual to pry loose from a company literature pertaining to its product. In the early days even an engineer working on a particular project practically had to sign his life away in order to be the recipient of an "instruction book."

c. Some manufacturers, however, do not seem to recognize the importance of issuing service bulletins concerning the installation, operation, maintenance, or overhaul of their engines. These service bulletins are considered as amendments to the particular "instruction books" to which they apply and since the "instruction books" are considered as part of the necessary data for certificating engines, naturally suitable service bulletins should also be edited and distributed in order to keep the instruction books up to date. The manufacturer considers his guarantee voided if his recommendations set forth in the respective instruction books are not followed. It is interesting to note, contrary to common opinion, that the manufacturers who distribute the most service bulletins (over 400 in several cases) have well-established positions in the aircraft engine field. Some manufacturers seem to feel that the issuance of service bulletins represents that an inferior article is being manufactured. This is an entirely erroneous viewpoint since dissemination of any literature represents "good will" and is so accepted by the industry. A few of the subjects that have been described successfully through the medium of service bulletins are listed as follows, although the various subjects that may be described are numerous:

1. Oversizing of parts.
2. Fits and clearances.
3. Cleaning of engine parts.
4. Assembly methods.
5. Availability of improved parts.
6. Engine oils approved.
7. Spark plugs approved.
8. Sand blasting.
9. Special inspection of parts, and so on.
10. Modernization and conversion of engines from one model to another.

Reference to Tables 4 and 5 will reveal the letter ("M") opposite some of the figures. This (M) indicates that, as the result of some chronic trouble, it was necessary for the airplane or airplane engine manufacturer or the Civil Aeronautics Administration to issue service bulletins or Airworthiness Maintenance Bulletins, respectively, to get this particular problem under control. There are hundreds of other instances on record in which service difficulties were quickly brought under control subsequent to the release of either service bulletins or Airworthiness Maintenance Bulletins.

d. The general public and all others, including pilots and operators of aircraft, inspection personnel, both manufacturing and governmental, manufacturers and all others who are closely associated with this fascinating industry, also have a problem on their hands. In fact, this problem might even be called a duty, that is, the necessity of applying rational thinking and treatment to problems which do arise and will continue to arise in the aviation industry.

The general public must realize that, although 100% safety is the ultimate goal, nevertheless, no form of transportation – mechanical or otherwise – has ever flashed across the horizon in which this Utopian desire has been

attained. It is believed that the safety record achieved by the airplane and component parts compares favorably with other forms of modern transportation.

The pilots, operators, and users of aircraft have the duty of cooperating with the airplane and airplane-engine and propeller manufacturers to the extent of observing their recommendations, which recommendations have been formulated usually as a result of a vast background of experience on the particular product discussed.

The manufacturers' inspection personnel has the responsibility of aiming continuously to achieve 100% inspection consistent with the economic factors involved.

The governmental inspection personnel covering civil aircraft must be painstaking in its daily endeavors to make sure that the minimum airworthiness requirements are always attained. They must judge things uniformly and impartially and must have the faculty of distinguishing between the important and unimportant details. This ability is an achievement in this particular case since so many unknown factors enter the picture, any one of which can make an item that appears relatively minor assume major importance by a peculiar sequence of events. It is almost necessary for an inspector to be a past master of "combinations" and "permutations" when a consideration of all the variables is made.

5. Replacement Parts

The general problem of engine replacement parts has been given considerable study in the past. The policy on this matter has been clearly expressed in the Civil Air Regulations. However, it is generally known that there are a number of unapproved replacement parts available for use in certificated engines. A number of the aircraft engine companies have recognized and have adopted the practice of distributing service bulletins warning against the installation of any non-genuine or unapproved parts in any of their engines, at the same time stating specifically that such practice causes their engines to cease being the designated engine and that the engine guarantee immediately becomes null and void.

The regulations provide a means to prevent the usage of structural parts of this nature in engines installed in airplanes issued aircraft certificates. An owner, to keep his aircraft certificate in force, must maintain his engine by subjecting it to overhaul when necessary or at reasonable periods of time so that the engine is at all times airworthy. At the time of each overhaul the manufacturer or mechanic is required to furnish the owner with an affidavit showing that the structural replacement parts used are approved. The affidavit must be pasted in the engine log book at each overhaul.

The term "structural engine parts" is interpreted as referring to all stressed moving and major parts of an engine as well as any minor parts of special construction which might cause a forced landing upon failure. The term does not include standard commercial parts of conventional materials which can be compared satisfactorily with the original part such as minor bearings, gaskets, minor studs, bolts, nuts, pins, safety wires, and so on. However, due to the special materials and fabricating methods used for parts of engines of any horsepower, all engines have very few parts of those mentioned which are not classified as structural engine parts, either by us or by the engine manufacturers. Therefore, the engine manufacturer's opinion must be obtained regarding these parts.

All parts purchased from the original manufacturer or his authorized service agency may be considered as approved. Under the terms of his engine type certificate, a manufacturer is only permitted to produce parts for service use in conformity with his sealed drawing list. Serviceable parts or parts purchased from other sources must bear the original manufacturer's marking and identification to be considered as approved.

Replacement parts will be approved, provided an engine of the specific type for which approval of these parts is desired has been equipped with the specific parts and has satisfactorily completed endurance tests as prescribed in Civil Air Regulations, Part 13. Therefore, it will be necessary for engine replacement parts manufacturers to equip an engine with all of the parts which they wish approved for the engine and run it either 50 hr full throttle at the rated speed or the complete 150 hr of testing, depending upon the circumstances involved. The engine may be mounted on any type stand with ordinary airplane instruments provided all the applicable readings required by Civil Air Regulations, Part 13, are obtained every 30 min. An inspector will be supplied to witness the test and to examine the parts before and after the test, as well as to make measurements of the parts.

In addition, in accordance with Civil Air Regulations, Part 13, it will be necessary to submit technical data suitably describing the parts as enumerated therein.

Approval of the replacement parts is then contingent upon the submission of a satisfactory report and our receipt of a satisfactory factory inspection report concerning the manufacturer's facilities and his supply sources. If approved replacement parts do not perform satisfactorily in service, the approval granted to the replacement part manufacturer will be revoked.

Apropos of this subject, it is known, in general, that the aircraft-engine manufacturer, not unlike the automobile manufacturer, is unsympathetic toward the use of so-called "pirate parts" in his product. His engine is the result of a coordinated organization composed of engineering, metallurgical, manufacturing, inspection, service, and sales departments. Whatever prestige his product has acquired is due to the efforts of this organization in both a financial and material way. Many hours of developmental testing have been run on each model prior to the submission of the engine for airworthiness approval. Moreover, a great deal of experimental testing is constantly in progress to improve the reliability and service life of each model. Progressive engine companies conduct a continual investigation of a large number of materials and accessories to determine which may be used with each model of their aircraft engines without impairing or adversely affecting the proper functioning or reliability of the engine. Certain materials and accessories thus investigated are released by these companies for use with their engines. Such a release signifies that the specified material or accessory may be used with the specified engine model without thereby voiding the engine guarantee. The use of materials and accessories not released constitutes a misuse of the engine and voids the guarantee. The experience of service operation, both commercial and military, is likewise available to the engine manufacturer. He is also in close touch with the latest metallurgical research of the steel and aluminum industries in addition to carrying on his own metallurgical activities. The major aircraft-engine companies manufacture at least 90% of their engines exclusive of accessories.

Extreme care is given to the material, the manufacture, and the quality of each part. When established for replacement, these parts are priced on manufacturing cost plus overhead and a fair profit. The operator is then assured of proven airworthiness and quality when he procures replacement parts from the original engine manufacturer.

The independent maker of replacement parts, on the other hand, enjoys few, if any, of the foregoing facilities. Many of them may be motivated by the desire for profit with minimum responsibility and many times, with no reputation at stake, the independent manufacturer can compete quite easily on a price basis with parts which may prove inferior. Despite the fact that he is required to identify such replacement parts as of his own manufacture, the reputation of the independent maker does not suffer greatly by failure of the parts in service. In the eyes of the public it is always the original engine manufacturer whose product is at fault, regardless of the conditions incident to the failure. Furthermore, unless the vendor has an inspection system of an unusually high order, it is not unreasonable to assume that the quality of his production parts may not measure up to the standard of the sample parts that have been submitted for dimensional inspection and to a competent testing laboratory for certified material analysis. Uniformity of heat-treatment is possible only by constant and close inspection supervision and, in the interests of economy, the vendor may easily overlook this most important item.

A 300-hr flight test of one engine in an airplane is not indicative of the resistance to fatigue failure of the parts under test. Weaknesses of this nature are brought to light only by continuous testing on a number of engines and by coordination of service experience of various operators under flight conditions. This type of experience the independent manufacturer of replacement parts cannot afford to buy.

There is a general opinion that considerable demand for available replacement parts emanates not from the air carriers nor the private operator of some means, but from owners of small airplanes powered by foreign aircraft engines or by obsolete engines. In cases of this kind it is difficult for the owners to obtain approved parts. There is a general consensus of opinion that, if the replacement parts for these engines had to be approved by the method outlined for approving parts of engines still in production, the expenditure necessary for such approval would be prohibitive.

Four typical cases applying to engines in the foregoing category are, namely: (1) a replacement Walters piston pin; (2) a Salmson piston; (3) a Siemens-Halske knuckle pin; and (4) a Szekely piston.

In reference to replacement parts as just described, the following modified procedure would be generally applicable as a financially practical and scientifically adequate means of testing replacement parts for foreign aircraft engines or for obsolete engines of the types mentioned:

1. The submission to the Civil Aeronautics Administration of comparative cross-sectional area diagrams, showing the contours and thicknesses of both the replacement part and the original part, giving all pertinent dimensions.

2. Comparative microscopic examination of crystalline structure by a recognized metallurgist, and favorable comment from him in writing to the effect that the replacement part is at least equal to the original in crystalline structure.

3. Brinell, Rockwell and/or other hardness comparative tests sufficient to demonstrate that the replacement part is at least equal to the original.

4. Comparative physical breakdown test, demonstrating that the replacement part is at least equal to the original.

5. Homogeneity test (X-ray) to show that the replacement part is homogeneous in structure, in that there are no shrink cavities or other inconsistencies. This test should be performed periodically in the production of the parts.

6. Comparative surface flaw tests (magnaflux).

7. Comparative friction coefficient test, using a standard metallurgical laboratory method for obtaining comparative frictional measurements.

8. Comparative expansion coefficient tests, whereby comparative micrometer readings are taken at low- and high temperature points.

9. A statement of chemical contents and their proportions.

10. It may be necessary that some of the foregoing tests be performed under the supervision of Civil Aeronautics inspectors, their inspection to include an investigation into the factory facilities in order that a judgment may be made in regard to the ability of such factory to produce the part or parts under consideration.

6. Fuel and Oil

In order to protect their engines, most engine manufacturers have adopted the policy of establishing fuel and oil specifications to which all fuels and oils, respectively, must conform if they are to be used in the respective manufacturers' engines. These specifications may either be established from laboratory tests or correspond to the characteristics of a particular fuel or oil with which the engine successfully completed the prescribed tests. In any event, all manufacturers specify at least the minimum octane aviation fuel which may be used and the generally recommended brands, grades, and viscosity of the lubricating oil that should be used.

The manufacturers of oils approved under their specification are then held responsible for the quality of their product. The introduction of any substance into an engine, into the engine oil, or into the engine oil system, which has not been approved by the company cancels the approval of the oil used in that engine.

It is generally known that considerable difficulty is being experienced due to cracked piston and cylinder heads, rings and valves sticking, and poor idling characteristics where operators use automobile gasoline in their aircraft engines. At present no aircraft engines are certificated for operation on automobile fuels. Therefore, the operator using such fuels must accept the responsibility for unsatisfactory conditions resulting therefrom, especially when the engine is damaged or a serious accident occurs.

There is a vast difference between the characteristics of aviation and automotive fuels which is really a subject in itself. The higher operating temperatures of the aircraft engines appear to accelerate the formations of gum and carbon when the cheaper grades of cracked fuels found in some automotive fuels are burned in them, thereby creating a possibility of malfunctioning of the rings and valves.

The most serious condition is the possibility of the occurrence of vapor lock due to the use of automotive fuels in aircraft fuel systems which operate at higher temperatures, especially when flying under high atmospheric temperature conditions. The specification limits for vapor

pressure are 7 psi on aviation fuels and 11 psi for automotive fuels. Therefore, it is apparent that the latter would boil much more readily, thereby creating vapor and consequent stoppage of fuel flow more readily. This difference in vapor pressure may affect the critical altitude of the aircraft as much as 10,000 ft.

It is known that some operators are using additives or special compounds which are claimed to promote better lubricating qualities of the crankcase oil. Certain demonstrations have been evolved which indicate an improvement in the breakdown characteristics of the oil under pressure. However, it would seem that this would not indicate the functioning of the compound under operating conditions. During operation of the engine the lubricating oil is subjected to some of the processes utilized during its refining such as temperature, pressure, evaporation, condensation, centrifugal force, and so on. However, the conditions in the refinery are accurately controlled. The crankcase oil is also subjected to contamination by carbon, moisture, acids, dirt, and other foreign matter. Therefore, any compound or blend used in the oil must not react chemically to form solutions which would cause deterioration of special bearing alloys. Some operators can substantiate increased service from the use of such products, while others can supply information as to dire results from its use. There is always the problem of whether such compounds will be of sufficient aid to warrant the extra expenditure.

Practically all standard brands of lubricants are blended and refined by organizations cognizant of operating conditions. As stated previously, the engine manufacturer furnishes a list of oils approved for use in his product and has done so after exhaustive tests to determine their suitability; therefore, it is a sound policy to follow his recommendations.

7. Detonation

The disturbing "ping ping" which all of us have heard at some time or other in our automobile engines, especially when going up a hill at low speed and with large throttle opening, is familiarly known as detonation.

Detonation has been defined as "the combustion phenomenon which results in an abnormal and uncontrollable rate of pressure rise which takes place with sufficient rapidity to set up waves within the combustion-chamber gases, which waves travel at, or greater than, the velocity of sound." It has been attributed to the almost instantaneous burning of the combustible mixture ahead of the normal flame front under certain conditions. For practical purposes, detonation may therefore be defined as a too rapid burning of a portion of the combustible mixture in the cylinder. The general indications of detonation, causes of detonation, results of detonation, and prevention of detonation are discussed briefly in the following:

(a) Although at present positive instrumentation which indicates detonation directly is not commercially available, the indirect indications that detonation is occurring are by engine roughness, cylinder-temperature increase, erratic fuel-air ratio, exhaust flame coloration, fluctuating manometer or manifold pressure readings, and generally decrease in power.

(b) The usual causes of detonation are the use of fuels of low octane value, low fuel-air ratio (leaning out), and engine operating conditions which cause an increase in the peak temperature in the combustion chamber, such as operation on one spark plug, operation at excessive power, and so on.

(c) The results of detonation are innumerable; however, there generally is a loss of power, overheating, preignition, and physical damage.

(d) Detonation can best be avoided by careful observance of specific engine operating instructions, by the use of the proper grade of fuel, and by maintaining the engine in proper mechanical operating conditions.

(e) Detonation usually can be stopped by reducing the manifold pressure, enriching the mixture, and reducing the carburetor air preheating to the minimum temperature at which icing of the carburetor may be prevented.

(f) In the event that detonation is caused by engine mechanical condition, check the ignition timing, the spark plugs, and see that the proper grades of fuel are used.

8. Quality Control

By "quality control" is meant the extensive and exhaustive system of inspection and tests to which it is necessary to subject all engine parts, both individually and collectively, to insure that the complete engine, when it is shipped out of the factory, is the type of equipment which will give satisfactory service through at least several overhaul periods. If failures occur with any degree of regularity, although the general design is satisfactory, you may rest assured that in some part of the inspection system there has been a breakdown in the quality control of the particular part that failed. Manufacturers who have a suitable inspection system, which normally requires that at least 10% of the workers actually are engaged in inspection, can readily handle additional inspection duties necessary to correct a breakdown in the inspection system, whereas manufacturers who are undermanned in this department may remedy one trouble only to have the general system fail in some other spot as a result of lack of inspection personnel.

It is felt that close to 50% of all the engine structural failures described on Table 4 are the results of a breakdown of the quality control system. Failures from surface defects comprise a good portion of the failures. Innumerable cases are known of crankshafts cracking owing to fatigue initiated by galling.

The following inspections listed give a brief idea of some, but not all, necessary inspections that must be made:

(a) Tools, fixtures, gages, and so on.

(b) Material, that is, forgings, bar stock, fuels, and so on.

(c) Purchased materials, that is, valves, piston rings, and so on.

(d) Heat-treat.

(e) Final inspection.

Special apparatus, such as microscopes, tensile and endurance testing machines, X-ray machine, spectrograph, oil laboratory, chemical laboratory, and so on, are familiar sights in most of the companies.

Some of the parts, such as crankshafts, may have as many as several hundred inspection operations. Many large companies have such signs as "Avoid Sharp Corners," "Provide Radius," and so on, plastered all over the drafting room and machine shops as a daily reminder that these precautions should be followed at all times. The inspectors are cautioned to observe that elimination of sharp corners is practiced religiously. Tool marks are not permitted on highly stressed parts in stressed areas.

Now practically every reliable engine manufacturing

company has magnaflux equipment to detect material defects. Magnaflux is a magnetic method of recent development used as an inspection tool for detecting imperfections, both detrimental and non-detrimental, in magnetic steel. This is a non-destructive method of inspection. This method must be used with discretion since it is not employed for the purpose of rejecting all parts that show indications, but to eliminate all parts that may cause failures. In many companies practically all steel parts are magnafluxed and inspected for defects.

Cracks are the most dangerous defects and their detection is no longer guesswork when magnaflux equipment is available.

The engine parts are assembled following many subassembly fits, clearances, checks, and sundry other inspection details. The engine is then tested, first being given a production test during which various performance checks are made. Teardown and inspection then follow, followed by final assembly and final test. If everything is satisfactory up to now, the engine is packed and shipped; otherwise, the necessary corrections must be made before the engine is released.

Good quality control insures a high standard of quality, satisfactory performance, and satisfied customers, which features, when obtained and when other things are equal, insure success in this highly competitive industry.

9. Overhaul

The owner of an aircraft, to keep his aircraft certificate in force, must maintain his engine by subjecting it to overhaul, when necessary or at reasonable periods, so that the engine is at all times airworthy. Although the Civil Aeronautics Administration does not prescribe definite periods for the overhaul of each approved engine, it places the burden upon the owner either to overhaul the engine at the manufacturer's recommended period, or to demonstrate that the engine is not in need of an overhaul. This policy is enforced by our inspectors refusing to re-certificate aircraft with engines found to be in unsatisfactory condition.

Reference to the manufacturer's handbooks covering the various engines will indicate that there often is no definite opinion regarding overhaul periods and, in many instances, it is stated that engine overhaul should be carried out at the discretion of the operators.

Although it is generally known that aircraft-engine manufacturers usually recommend from 400- to 600-hr major overhauls; nevertheless, it is suggested to owners and operators that they obtain the latest recommended procedure from the pertinent engine manufacturer for their particular operation. The engine manufacturers accumulate a manifold service experience background as a result of their close contact with the operation of their engines.

As intimated, there are no hard and fast rules regarding engine overhaul time, particularly the maximum allowable, because of the many variables which must be considered. Manufacturers, in general, have made a practice of taking each individual operator and considering the particular problem and record. All manufacturers generally stress starting off conservatively as regards overhaul periods on new engines or new equipment. Even when an operator takes over equipment which is new to his particular operation, the manufacturers usually recommend that increases in overhaul periods be approached gradually, judging by the condition of his engines at time of overhaul and experience of the operators.

A certain group of engines used in the Civilian Pilot Training Program undergoes major overhaul every 800 hr, the subsequent inspections showing the engines to be in very good condition. Engines of this type in private practice normally are overhauled every 500 hr as a matter of necessity. This example demonstrates the value of operation under controlled maintenance conditions similar to that practiced by the airline operators.

After the overhaul, it is necessary satisfactorily to run in and test the overhauled engines. There exists a great divergence of opinion and practice in the matter of "running in" and testing aero engines after complete overhaul, and so it is deemed to be desirable to lay down a few basic principles and a minimum standard which must be observed in order to insure airworthiness.

The running in and test of an airplane engine following assembly is an integral part of the overhaul procedure; therefore, the operation must be done in accordance with the procedure as outlined by the respective manufacturer.

There are two distinct objectives to be achieved: first, the settling of the engine parts into their proper working order, which is effected by a careful and thorough running in at light load and low speed; second, to determine whether the engine performance will meet the minimum rating for its type and model and is of a satisfactory quality.

It may be of interest to mention that some of the engine manufacturers and service stations are cooperating with operators in the Civilian Pilot Training Program to the extent of furnishing such operators with a loan engine during the time their unit is being overhauled; such overhaul normally requires four or five days.

10. Starting

A good many of you probably had automobiles before the advent of one of Charles Kettering's "brainstorms," the starter. If so, you recall that on cold days your car invariably stayed in the "barn," or, if you were of an intrepid nature, you paid the price of being obstinate and actually started the car (sometimes) by having a sore arm at the upper end or, at least, sore fingers. All of this usually was accompanied by a long series of expletives that probably would make some of us timid souls blush to the roots of our hair.

The man's world of automobiling was changed when the automatic starter was introduced, and it was not long before women in scores invaded what man considered his private domain and bumped fenders with him in all seriousness and with full presence and intent.

We are at that same impasse now, figuratively speaking, for the automatic aircraft engine starter is here and women are casting glances high in the air to meet man's challenge of high flying tactics, even at the price of donning a pair of wings to soar into the dizzy heights.

Cold-weather starting is an accomplishment inasmuch as, under cold-weather conditions, although the engine fires from the original prime, it fails to continue running due to the exhaustion of fuel before the engine picks up the carburetor charges. During this interval, which varies on different engines sometimes being at least several minutes, fuel must be fed to the engine by means of the engine primer. To those of you who doubt this statement, just check your automobile starting technique in which the choke virtually serves as the priming actuating means.

It is generally considered good practice when operating

in temperatures below 20 F, to warm the oil before starting the engine. Frankly, it is believed that an operator of a fleet of aircraft would actually be ahead economically if he made it a practice that all his aircraft engines had their oil warmed up before starting if the air temperature is below 50 F. Heating the oil can be accomplished easily by draining the oil when flying is stopped for the day and warming it preparatory to starting the engine again.

In any event, if the engine has been standing any length of time, it is always recommended in any kind of weather, hot or cold, to turn the engine over by hand at least two times to obviate the danger of blowing off cylinder heads as a consequence of oil or fuel inadvertently having entered the cylinder-head compartment. If the weather is cold, several extra turns will pay a premium in requiring less torque to break the parts away from the congealed oil.

Good maintenance men know that, if they are easy on engines during starting and warm-up, the engine will last much longer. There is no difference between an engine and a human body when it comes to starting technique. Those who think otherwise should familiarize themselves with the starting programs which all champions in the various "sports" adopt to warm up the system for the eventual "take-off" for fame and glory.

11. Personnel

Reference to Tables 4, 5, and 6 will reveal there are many failures attributable to personnel errors. Of course, it is recognized if you cared to press the point, that all the failures listed under any classification are personnel failures. It is granted that this would make a lovely grouping, but it covers too much territory; therefore, the breakdown system has been devised.

The types of failures listed under personnel in the charts are failures resulting from fuel exhaustion, fuel valve closed, improper lubrication, spark retarded, loose sparkplug wires, improper maintenance, automobile fuel, insufficient warm up, and so on.

Practically all of the structural failures may be considered personnel failures when it is considered that poor design and practices, faulty inspection, or failure of parts resulting from improper operation or maintenance revert directly to the personnel issue.

Let us consider the pilot because he also comes into this picture. He has many functions among which are: (a) operation of the engine controls; (b) interpreting of the readings; (c) maintaining operating limits in accordance with recommendations; (d) attempting to maintain powerplant efficiency at its peak, and so on. If he overspeeds the engine, overheats it, or uses too lean a mixture, incipient failure is likely to have started which will only manifest itself many hours later, at which time the cause for the failure has been lost in antiquity.

Now we can consider the maintenance angle. If an engine is in un-airworthy condition and no overhaul is conducted on the engine, failures will occur sooner or later if nature is running true to form. Juvenal's saying: "Nature and wisdom always say the same," is rather to the point. If an overhaul is conducted and it is done improperly, a failure is imminent.

Just a word in regard to quality control. Even if the best quality control is practiced, nevertheless if the parts are not demagnetized (a necessary precaution) following magnafluxing, there is the possibility of steel chips adhering to these parts and causing engine trouble.

What is the answer to this personnel problem, a veritable paradox from which there seems to be no escape? There is an answer to this problem, as well as every other problem which does or may come up. Although vaguely hinted at, it is rather delicately described under Item 4, "Education."

It is believed that the only way of accomplishing the objective, namely, eliminating the personnel errors, will be by a continuous collaborative and coordinated effort on the part of the industry, schools, and the governmental agencies to provide a virtual barrage of educational literature of high standard to all those engaged in this fascinating and swift-moving drama of speed, whether they be manufacturer, pilot, engineer, mechanic, office boy, or sweeper. Barrage is stated and barrage is meant because, if unhealthy ideologies can be inculcated into the masses by this technique, there is no reason why the same process cannot be efficiently used to indoctrinate all of us exposed to healthy literature with its ultimate purpose – improvement. This desire, once deeply injected into all of us, will fertilize and bear fruit in the form of its actual achievement.

As Fuller said: "If you have knowledge, let others light their candles at it."

12. Vibration

Although vibration has been a problem of engine designers from the very beginning, the aircraft-engine builders have only extensively studied the phenomenon of vibration during the last decade. First the large engine builders as a matter of necessity were forced to adopt means to control this disturbing element or the engines would suffer ultimate destruction if the small but not insignificant forces were not adequately controlled and harnessed. Now even the "Little Three" engines are subjected to vibration studies to keep away from what the layman is usually given to understand as "engine roughness." The principal reason for keeping away from excessive vibration is to prevent premature failures of the parts before their normal service life, thereby extending the service life of the complete engine and the related parts attached thereto, even of the aircraft itself. Of course there is this thing called "roughness" which, although very objectionable to operators of aircraft, nevertheless has not been the reason for the interest the aircraft manufacturers are putting into this phase of the work. But some day operators will demand the same freedom from disturbing vibration forces which freedom is now generally enjoyed by automobile owners.

The crankshaft used in an aircraft engine is subjected to a multitude of forces and vibrations. All of these conditions cannot be predicted accurately by the engine designer in his original analysis. Furthermore, other conditions, such as the engine mounting and method of operating the engine, enter into the picture. To date these conditions have not been very critical in the "Little-Three" engines, but other factors, such as increase of power output and speed, longer crankshafts, and developments such as use of metal propellers, and so on, will naturally change the present tranquillity as associated with these engines.

Torsional vibration of the crankshaft is due mainly to the firing impulses which cause displacement of large masses of the rotating system with respect to each other, actually twisting the crankshaft. Deflecting and restoring forces are set up which will cause failure if the displacements are large, or if they occur at resonance. Resonance may be defined as a phenomenon which occurs when the

frequency of a vibration-exciting force equals the natural frequency in the system to which the force is applied. Resonance results in vibration of great magnitude in the system.

Apropos of the problem of vibration, Part 13.66 "Vibration Stresses," of Civil Aeronautics Manual 13, is quoted:

With the increase of size, and consequent increase in power of the modern aircraft engine, the analysis of vibration problems for the purpose of controlling the vibration and corresponding stresses to safe values becomes more and more important. Problems of great practical significance, such as the balancing of machines, the torsional vibration of shafts and of geared systems, the vibration of impellers, the whirling of rotating shafts, the vibration of crankcases and adjacent parts, can be thoroughly understood only on the basis of the theory of vibration. Only by using this theory can a modern aircraft engine be constructed in which the working conditions of the engine are free from the critical conditions at which severe vibrations may occur. There are a number of textbooks in which the fundamentals of the theory of vibration are developed and their application to the solution of technical problems is illustrated by various examples taken in many cases from actual experience with vibration of machines and structures in service. The design of parts to withstand vibratory loads can be assisted by checking the theories developed by actual measurements of the vibration stresses in the parts. Stress pickups and suitable instrumentation are used for this purpose where it is possible to locate the present types of pickups on engine parts under operation.

It has also been demonstrated in practical fashion that vibration in the aircraft is greatly influenced by the design of the engine mounting and that it is possible, by proper elastic support, virtually to isolate all the engine sources of vibration from the aircraft structure without radical recourse from the present mount structural design. A study of the vibration characteristics of an engine-propeller installation consists of: (1) a determination of the frequencies of all the modes of vibration; (2) a study of the exciting forces; (3) the operating conditions. The engineer can then predict from these data the vibration characteristics of the installation. On the basis of these predictions, recommendations may be made regarding the avoidance of operation in certain troublesome ranges.

Considerable study has been made regarding propeller vibration. One method of conducting a propeller vibration study in flight is described as follows:

Vibrations which occur in a metal propeller under flight conditions are determined in tests by mounting automatic equipment consisting of a number of resistance pickups, batteries, amplifier, oscillograph, and collector rings, in the plane. Impulses recorded on the film in the oscillograph permit engineers to measure the vibration, determine its seriousness and, in many cases, locate the source from the frequency and characteristics of the vibration lines on the film.

Laboratory tests include suspending the propeller in an elastic sling and vibrating it under static conditions, use of propeller test rigs on which electric motors are used to whirl test new propellers, and test stands on which the experimental engine-propeller combination can be mounted and tested.

In flight-testing equipment, resistance pickups, carbon or metal strips ⅞ in. or more in length, are cemented on the propeller blades at the points where the stress is to be determined. Vibration in the propeller changes the linear measurement and the electrical resistance of the pickups. Current passes through these resistors from batteries, through the oscillograph and amplifier. Normal slight vibration which causes a fluctuation of electric current is recorded in a regular shallow wavy line. If abnormal vibration develops, the line becomes a jagged series of peaks which increases in size as the vibration increases.

Very little study has been given to vibration in wood propellers in view of the fact that the service record of wood propellers has been excellent. However, it is my opinion that wood propellers will eventually be analyzed as closely as metal propellers from the vibration standpoint.

In concluding, it is thought that the best available means for determining and controlling the vibration characteristics is by the use of sensitive electrical seismographs under conditions of actual operation.

13. Noise

There seems to be a matter-of-fact acceptance by many that aircraft and noise are virtually synonymous. A prevalent opinion seems to be that, if the engine noises were eliminated, the propeller noises alone would be more objectionable than when accompanied in a duet by the engine noises. It is going to be a hard problem to pierce the armor plate of such established opinions; nevertheless, I am glad there are some, who by the expression of their developments, have other opinions. It is fortunate that the airplane designers did not have the same prevalent opinion; otherwise, nothing would ever have been done to develop soundproof cabins (probably in self defense) which, although not completely sound proof, nevertheless have been very successful in damping out most of the engine and propeller noise. In addition, much work has been done on airplane ventilating systems, and so on.

Let us consider the sources from whence the noises originate and tackle them individually, that is: (a) exhaust, (b) pistons, (c) gears, (d) tappets, (e) clearances, (f) cowling, (g) mounts, and (h) propellers.

(a) The exhaust is as easily muffled as the exhaust in your automobile engine. There are many applications of exhaust muffling in the "Little-Three" equipped airplanes. Of course, mufflers should only be included as equipment after it has been demonstrated that airworthiness is not adversely affected. Due consideration must be given to the back pressure, fire hazard, installation, and structural requirements. In addition, it is necessary that the engine manufacturer's recommendations regarding his engine are considered.

(b) The piston noise is also a matter of the past, since some of the "Little-Three" engines are equipped with pistons which have approximately 0.001-in. instead of 0.004-in. clearance per inch of piston diameter and they are aluminum pistons too. This decrease in clearance cuts down piston slap, especially during warming-up operations.

(c) The gear noise is a tougher problem, but here too, some of the "Little-Three" engines are equipped with composition gears which reduce gear noise. Much work has to be done here, but the start has been made and it is encouraging. Other quieting means may be obtained by helical gears, spring-loaded gears which are always in contact, more carefully controlled clearances, and so on.

(d) The tappet noise has also been solved by the application of hydraulic valve lifters. This type of lifter has been used extensively in automobiles to take up, automatically, all clearance in the valve gear under all conditions of load, speed, temperature, and wear so as to give zero tappet clearance.

(e) The other clearances in the engine may be more rigidly controlled to prevent slapping and pounding. These noises are not considered to be excessive even as they are at present, so possibly no modification will be necessary.

(f) The cowling, of course, must be designed properly

and rigidly supported so that vibration of the various parts of the cowling and of the baffles does not result in drum beats, which probably occurs in the current aircraft but is unknown to anyone because this noise is comparatively muffled now.

(g) The mounts also should be designed so that vibration is minimized and also should incorporate proper elastic support at the mounting lugs, or other means to preclude transmission of vibration through these parts and, consequently, to the aircraft structure.

(h) It is recognized that propeller noise is a tough problem to solve. It is also recognized that it never will be solved if this opinion is accepted as final and no attempt is made to dispel the myth that propeller noise cannot be eliminated. This problem should be put in the laps of the acoustical experts who first of all will analyze the problem. When it is analyzed accurately, engineers should be able to apply various methods of attack to effect the desired results. Therefore, first obtain an accurate analysis of this problem subsequent to which let the engineers loose to solve it.

This problem is aggravating, but for the sake of our nerves, let's tackle it properly and solve it.

14. Standardization

The Civil Aeronautics Administration heartily approves the process of standardizing as applied to aircraft engines. This may be amplified by quoting parts 13.401, 13.404, and 13.931-n of the proposed Civil Aeronautics Manual 13 which is expected to be released soon in slightly modified form than when last presented to the industry for comment.

13.401 Materials

The structural parts of an engine should be made of materials which experience or conclusive tests have proved to be uniform in quality and strength or to be otherwise suitable for engine construction. It is suggested that the practice of referring to the Society of Automotive Engineers Aeronautical Material Specifications be followed. The Society of Automotive Engineers has prepared specifications for all classes of material used in aircraft, their engines, and major accessory equipment. In general, there are no limitations with respect to the material, provided it is suitable for the purpose and provided the material variation is within such limits as to permit satisfactory duplication of all parts of the test engine. Where practicable, metal or other fireproof materials should be utilized in engine construction.

13.404 Standards

Deviations from the existing standards should be established with respect to their safety. Technical comparisons or testing may be necessary to accomplish this. Army, Navy, SAE or any other such standards are particularly applicable to propeller shaft ends, hub fronts and rear cones, hub retaining nuts, engine nose plates, mounting flanges, nuts, bolts and material specifications. Some recommended practices are described in CAM 13.93. Refer to CAM 13.401 regarding recommended practices relative to SAE Aeronautical Material Specifications.

13.931-n SAE Recommended Practices Charts – Figs. 22-27. In accordance with CAM 13.404, a group of drawings describing the SAE recommended practices applicable to propeller shaft ends, hub fronts and rear cones, etc., are included. These practices are generally followed by manufacturers and are included herein so that ready reference may be made thereto.

The time may not be too distant when the standardization and supply by aircraft engine manufacturers, even by the "Little-Three," of complete power units ready to install in the aircraft becomes a reality. From the economic standpoint, much can be said for such standardization. From the practical standpoint, it seems as if the engine manufacturer should be able to do this very nicely, at the same time giving his engine the well-known "break" which, we usually understand from him, his engines do not get as normally installed. This would require much cooperation between the affected parties for its consummation and probably is suggesting a too radical method of doing things, but the idea is sound.

15. Visibility

This is recognized as an aircraft problem and is going to be considered as such. It is only being touched upon in this paper in view of the fact that, if ultimately the public demands improved visibility from the aircraft manufacturer, the latter in turn probably will be forced to place certain limitations on the engines which will equip his aircraft. The aircraft-engine manufacturer will, accordingly, be obliged to design engines to specific specifications. To meet these specifications the "Little-Three" engines of the future may be of much different design. If retained in the front of the fuselage, the design may be an inverted V of 4, 6, or 8 cylinders. These hypothetical engines would not compare with the current engines from an economical and weight basis, but the best all-round aircraft incorporate many compromised designs to give the best overall efficiency. The public will demand many things in the future and vision surely will be the one which will be demanded most of all. The mass demand of the public has usually been stopped only at the price of acquiescence.

16. Welding

This subject is being discussed because of the large number of inquiries received by the Civil Aeronautics Administration regarding welding on "Little-Three" engines.

The pertinent regulation covering welding is contained in Civil Air Regulations 18.7225 which states: "Welding shall not be done on any structural part of a certificated engine except in special cases when it is proved conclusively to the administrator that the repaired part is as airworthy as originally."

The term "structural engine part" refers to the major parts of an engine and any minor parts of special construction which might cause a forced landing if they failed. All highly stressed parts, including all reciprocating and rotating parts, and also crankcases, are included in this category.

This attitude is based on the fact that engine designs, in general, incorporate practically no welded parts of any material due to the vibration loads which are present. Cast-aluminum alloys are considered more difficult to weld than other engine material so engine manufacturers do not do any welding of these alloys in their production except to add material to non-stressed parts such as repairing foundry damage to cylinder-head fins and repairs of this nature.

The cast aluminum used in engines is of high-strength heat-treated alloys. Any welding is likely to destroy the properties of adjacent material even though heat-treatment is applied before and after in an effort to restore the part to uniform strength again. Aircraft-engine cast-aluminum parts are highly stressed and, due to their thin sections, it is very difficult to produce a good weld. All evidence, so far, has indicated that there is always a possibility that the part is actually weakened after the welding repair, although it may be locally reinforced.

The Aluminum Co. of America tacitly refrains from committing itself on the advisability of welding aircraft engine aluminum parts that are under load during operation.

In any event, if a case is considered special, the manufacturer of the affected engine is contacted to aid the Civil Aeronautics Administration in arriving at a decision consistent with the facts at hand. The engine manufacturers have accumulated stores of information on their engines which is of invaluable assistance to the Civil Aeronautics Administration in cases of this nature.

17. Public Demand

A review of the progress exemplified by the "Little-Three" engines reflects the enormous part which public demand has played in their development.

It has truly been said that the public is wiser than the wisest critic. The public surely has demonstrated that the law of "supply-and-demand" works both ways.

The manufacturers should be given credit for the various developments effected and which they incorporated on the "Little-Three" engines during the last few years. They are given credit for that and also for the fact that they have kept their ears close to the ground and have felt the pulse of the public opinion and its twin, public demand, which precipitated practically every development.

Some of these developments, characteristics, or features (comparatively speaking) which are now incorporated on these engines as a result of the unofficial "Gallup polls" conducted are:

a. Higher horsepower.
b. Low weight per horsepower.
c. Dependability and long life.
d. Fuel injector.
e. Low costs.
f. Freedom from servicing or maintenance.
g. Smooth and flexible operation.
h. General design for coordination with airplane form.
i. Better cooling (tunnel).
j. Pressure baffling.
k. Suitable speeds to permit high propeller efficiency by means of reduction gearing.
l. Dynamic dampers.
m. low fuel and oil consumption.
n. High volumetric efficiency.
o. Dual ignition.
p. Wet oil sumps.
q. Quieting means by use of mufflers, close-fitting pistons, composition gears, and hydraulic valve lifters.
r. Automatic valve-gear lubrication.
s. Oil coolers.
t. Generators.
u. Starters.
v. Elastic mounts.
w. Fuel pumps.
x. Efficient carburetor heaters.
y. Simplified installations.
z. Adjustable pitch wood propellers.

A review of this list of items will reveal that the only thing which the "Little-Three" engines do not have which the big engines do have is a supercharger. The reason is that power can be obtained much easier by other methods on this type of engine.

Since neither the public nor the manufacturers are infallible, some of the items mentioned may "boomerang," in which case there will be a dissection and removal of this part from the powerplant, much to everyone's relief.

If we bend our ears to the ground, the following faint rumblings, which may be the forerunners of future developments, are heard:

a. Improvement in all the items just listed which may bear improvement.
b. Better visibility.
c. Better take-off performance.
d. Foolproof devices.
e. Reduction in noise.
f. Propeller gear-box reduction.
g. Metal propellers.
h. Controllable-pitch wood and metal propellers.
i. Vibration damping equipment.
j. Oil filters.
k. Air cleaners.
l. Automatic heat and mixture control for carburetors.
m. Automatic heat control for oil.
n. Intake silencers.

The rumblings are few and faint now, but will they be so if the following opinion of Frank A. Tichenor is realized:

> "In a few years the young man and even the young woman who has not learned to fly will be regarded as natural phenomena as today are those who cannot drive automobiles."

■ Conclusion

The panorama of the present-day problems in light airplane engines has been presented. I, personally, do not consider that the picture which has been unfolded should be received with dismay. On the contrary, this survey, far from indicting, really establishes the position of the "Little-Three" type of engine. It is hoped this will act as a stimulus for greater improvement and at the same time focus the spotlight of admiration on the "Little-Three" engines. Their present record of achievement which is truly remarkable can only be maintained by a herculean effort and surpassed by a miracle. You and the general public are the jury. Until you make your decision, the record is being inscribed on the scrolls of time, the arbiter which weighs all things with infinite justice.

■ Acknowledgments

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Bibliography

1. "Private Flying Operations," *Civil Aeronautics Journal,* Oct. 15, 1940.

2. Gaylord W. Newton: "A Survey of the Cause of All Powerplant Failures Reported to the Bureau of Air Commerce Between Jan. 1, 1931, and Dec. 31, 1935," Information Paper No. 8, Bureau of Air Commerce.

3. Gaylord W. Newton: "A Survey of Mechanical Failures of Aircraft During 1936-1937," presented at National Aircraft Production Meeting of the Society, Los Angeles, Calif., Oct. 13-15, 1938.

4. Airworthiness Maintenance Bulletin No. 41, Civil Aeronautics Administration.

5. Certificate and Inspection Division Release No. 36, Civil Aeronautics Administration.

6. Certificate and Inspection Division Release No. 21, Civil Aeronautics Administration.

7. Civil Aeronautics Manual No. 13, Civil Aeronautics Administration.

A Cetane-Number Study

THE object of this paper is to emphasize some of the problems entering into the manufacture, control, and utilization of diesel fuel for mobile and high-speed diesel engines. The importance of ignition quality, as expressed by cetane number, is illustrated for rating the performance characteristics of a fuel. The relationship of cetane number to such engine characteristics as smoke, roughness, carbon deposits, starting, and exhaust odor, is given.

The influence of the crude from which the diesel fuel is derived, and the influence of the method and degree of refining on the properties of the finished fuel, are brought out in an attempt to explain the cause for the variation in diesel fuels. The properties of straight-run and cracked fractions of gas oils from several crude sources are given, as well as the properties of several fuels obtained by blending these stocks.

The final plea in the paper asks for a single specification for a diesel fuel which will satisfy the refiner as well as meet the requirements of mobile and high-speed engines of present design.

THE modern high-speed diesel engine is definitely establishing a place for itself as a dependable, efficient, and economical source of power and, because of its increasingly good performance, it is offering important competition to its gasoline-burning contemporary. Its inherent advantages are being recognized rapidly and its use in transportation, industry, and military operations is expanding steadily. Modern mechanized warfare, with its dependence on tanks and trucks, provides a wide field for the diesel. If one considers the hazard to the crew "buttoned up" with many gallons of gasoline inside a turreted multiple-gun tank, the advantage of a non-volatile fuel is obvious. The introduction of the "Molotov Cocktail" emphasizes the importance of the safety of diesel fuel. In the desert warfare of Northern Africa where extremely hot temperatures were encountered, the diesel-powered trucks were said to be notably successful because of the reduction in fire hazard and their dependability of performance.

Inasmuch as fuel plays a major part in diesel performance, in this paper we shall make a study of the problems associated with the manufacture, control, and application of this fuel.

While the fuels demanded by the high-speed diesel engines of today sacrifice none of the advantages of safety or economy characterizing diesel fuels in general, yet the high-speed design does require a fuel which has been more carefully selected and more critically controlled if it is to meet all the requirements and give satisfactory operation. With the high-speed engine, certain of the fuel characteristics become especially important. One of these is that property known as ignition quality, which is the ability to initiate or start the burning process in the combustion chamber. As engine rpm is raised, the time interval between the injection of fuel into the combustion space and the start of burning becomes a greater proportion of the total time available. It is this time interval, between injection and start of burning, and which is called "delay," that forms one basis of measurement for rating the ignition quality of diesel fuels.

The proposed method of test for ignition quality of diesel fuel (see ASTM Report of Committee D-2, Appendix III for 1940) is a method for determining the ignition quality of diesel fuels in terms of an arbitrary scale of ASTM cetane numbers.

Table 1 – Properties of Primary Reference Fuels

	Cetane	Alpha-Methyl-Naphthalene
Specific Gravity, 60/60 F	0.776	1.025
Boiling Range, F	544.1–553	469.6
Freezing Point, F	61.5	−7.6
Iodine Number	Nil	....

Table 2 – Properties of Secondary Diesel Reference Fuels

	Shell High 1801-Batch 7	Shell Low 1802-Batch 1
Gravity, API deg	40.4	47.4
Flash, F	200	144
Fire, F	225	164
Viscosity at 100 F, SSU	37	36
Pour-Point, F	15	B-20
Sulfur, %	0.40	0.04
Carbon Residue, %	0.003	0.02
Ash, %	0.001	0.002
Aniline Point, F	174	191
Distillation		
Initial Boiling Point, F	384	392
10% Recovered, F	452	425
20% " "	479	432
30% " "	508	438
40% " "	529	444
50% " "	549	450
60% " "	567	457
70% " "	590	467
80% " "	614	485
90% " "	645	509
End Point, F	700+	577
Cetane Number	72.5	22.0

[This paper was presented at the National Fuels and Lubricants Meeting of the Society, Tulsa, Okla., Oct. 24, 1941.]

[1] Cetane can be purchased from the E. I. du Pont de Nemours & Co. for $35 per gal and alpha-methylnaphthalene can be purchased from the Reilly Tar and Chemical Co., for $6 per gal.

# of DIESEL FUELS

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The ASTM cetane number of a diesel fuel is defined by, and is numerically equal to, the percentage by volume of cetane in a mixture of cetane and alpha-methylnaphthalene which the fuel matches in ignition quality when compared by the procedure prescribed in this method. Thus, by definition, alpha-methylnaphthalene has a cetane number rating of zero, and cetane, of 100.

Cetane ($C_{16}H_{34}$) is a straight-chain hydrocarbon of the paraffin series. The ignition quality of this material may be duplicated readily in manufacture and also possesses satisfactory stability characteristics. Alpha-methylnaphthalene is a coal-tar product. The properties of these two primary reference fuels are given in Table 1.

Because of the higher cost of the primary reference fuels[1], secondary fuels are ordinarily employed for routine ratings. These secondary reference fuels at present are:

Shell secondary diesel fuel for high-reference use.

Commercial methylnaphthalene for low-reference use.

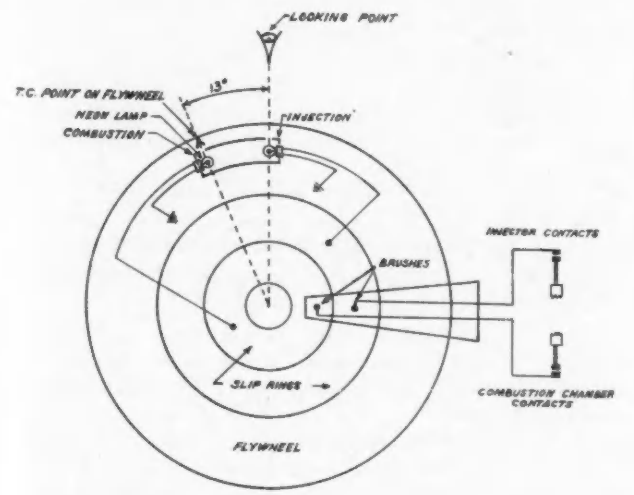
Shell secondary high-reference fuel is a straight-run gas oil having an ignition value of 72.5 cetane number. The other properties of this fuel are given in Table 2.

It has been advocated that commercial methylnaphthalene be replaced by a petroleum fuel having suitable properties. Test work on such a fuel has been completed and this material, having a cetane number of 22, is now available from the Shell Oil Co., Wood River, Ill. The properties of this petroleum low reference fuel are likewise given in Table 2.

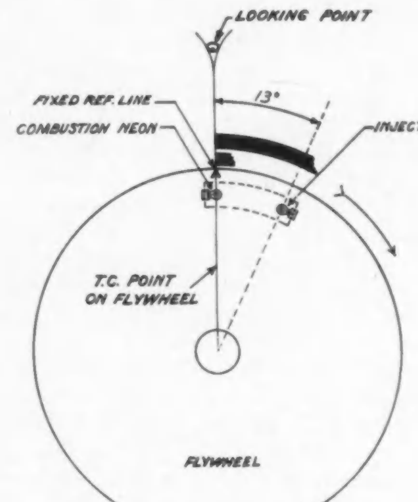
The test engine employed for determining the cetane rating of diesel fuels is an adaptation of the standard CFR gasoline rating unit. The engine is modified by the addition of a variable-compression diesel cylinder, injection equipment, and delay-measuring apparatus. A detailed description of this equipment is to be found in the ASTM Procedure.

Ignition-quality, or delay-method cetane-number determinations are made on an unknown fuel by running it in the testing engine and adjusting the compression ratio of the engine until combustion of the fuel occurs at top-center position and exactly 13 deg after the start of injection. By comparison with two blends of reference fuels of known cetane numbers, one of higher and one of lower number, the cetane number of the unknown fuel is ascertained by interpolation.

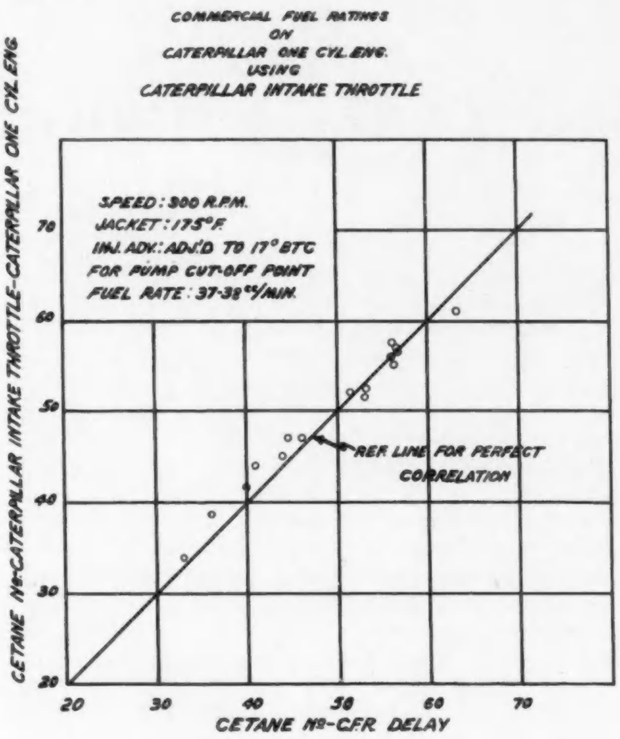
The apparatus for indicating the delay between the point when injection takes place and the start of combustion, or ignition, is rather simple. Two neon lights are mounted on the flywheel so as to rotate with it and are each connected to a slip ring to permit external connections. One of these neon lamps, which we will call the combustion indicator lamp, is mounted on the flywheel so that it is opposite a stationary "looking point" containing a hair line



■ Fig. 1 – Apparatus for indicating ignition delay



■ Fig. 2 – Condition in ignition-delay indicator with injection at 13 deg before top-center and combustion at top-center



■ Fig. 3 – Correlation of commercial diesel fuel cetane ratings obtained with Caterpillar cetane valve with those obtained by the fixed-delay method

and mounted on the engine, when the piston is in the top-center position. Hence, if the "looking point" is located at the top vertical position over the flywheel, then the combustion indicator lamp will be mounted on the flywheel in the same position as the top-center mark on the flywheel. This is the condition shown diagrammatically in Fig. 1.

The second neon lamp, which we will call the injection indicator lamp, is mounted on the flywheel 13 deg ahead, or before the combustion indicator lamp. This arrangement is also shown in Fig. 1.

The combustion indicator lamp is connected through the slip ring to a contact switch which is operated by the rise in pressure due to combustion. The injection indicator lamp is connected through its slip ring to a contact switch which is operated by the injection valve. Hence, when the injection pump is timed to inject fuel at 13 deg before top center, the operator, looking through the "looking point," will see the flash of the injection indicator lamp under the hair line. Further, if the engine compression ratio has been adjusted so that the fuel gives the arbitrarily fixed delay of 13 deg, then the operator will see a second flash which also appears under the hair line. Fig. 2 shows this condition. Persistence of vision makes the first flash appear to be beside the second flash. Now, if the fuel and engine conditions are such that the delay is only 10 deg of crank angle, the flash from the combustion indicator lamp will occur 3 deg before it reaches the hair line and will so reveal itself to the operator who then can readjust engine compression ratio to give the fixed delay of 13 deg.

Due to its simplicity and relatively low cost the coincident-flash method was presented to the ASTM and is the method now in general use. There are several other modifications of this method that may be used, though the principle of application of each is much the same.

The Penn State method is an outstanding one and utilizes magnetic pick-ups instead of mechanical contactors for determining the beginning of both injection and combustion and for operation of the indicating neon lights on the flywheel. Another method that is in an advanced state of development employs an ignition lag meter, similar to the knockmeter used in octane rating of gasoline, thus dispensing with the need for continual observation of the neon lights. An instrument of the meter type may be used in conjunction with either mechanical or magnetic pick-ups.

■ Caterpillar Cetane Valve

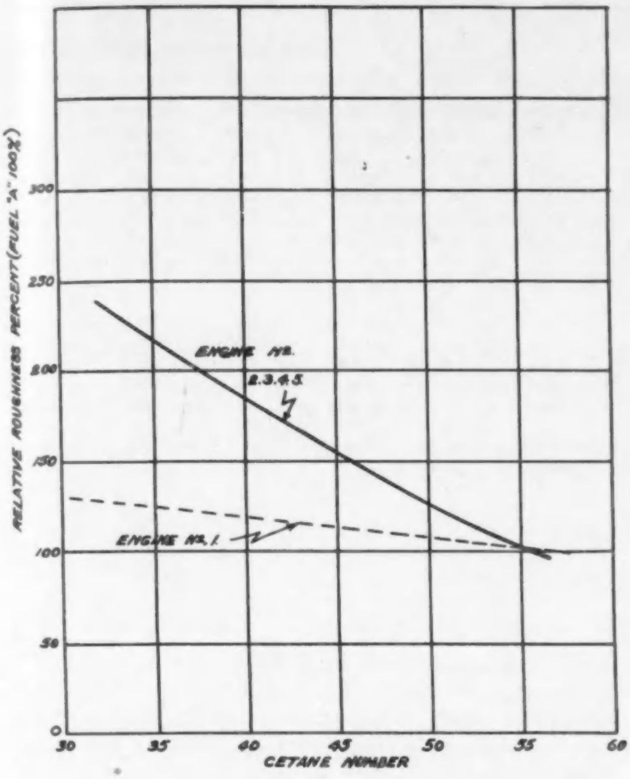
Another engine test method that shows promise for determining the ignition quality of diesel fuels, particularly when used on engines somewhat larger than the CFR engine, is the Caterpillar cetane valve. Developed by the Research Department of the Caterpillar Tractor Co., Peoria, Ill., this instrument, which is essentially a motor-driven plate valve of special design, is mounted in the air-intake opening of a one-cylinder test engine. Ratings are obtained by throttling the intake air to an extent where misfire of the fuel occurs. The ignition value for the corresponding misfire point is read directly from an attached scale calibrated in cetane units. The cetane scale is calibrated by rating the actual misfire points of various blends of cetane and alpha-methylnaphthalene. Checks of any point may be made as often as necessary either with primary or secondary reference-fuel blends.

Fig. 3 shows a correlation of fuel cetane ratings obtained with this valve and those obtained by the fixed-delay method. The engine on which this valve was mounted was a one-cylinder Caterpillar engine of 5¾-in. bore and 8-in. stroke, operating at 900 rpm. The results given were obtained on commercial diesel fuels selected at random. All of the points fall within 2 cetane numbers of the 45-deg correlation line. There is some indication that the cetane number as determined by the throttle valve will give higher ratings than the CFR delay method on lower value fuels and lower ratings on higher value fuels.

Numerous indices have been developed from time to time for indicating fuel ignition quality. These indices are based on physical or chemical properties of diesel fuels,

Table 3 – Variation of Other Indices with Cetane Number

	Cracked Wax	Shell Reference Fuel	Penn. Gas Oil	Mid-Cont. Gas Oil	Calif. Gas Oil	South Texas	Alkylate Distillate
Cetane No.	96.0	70.5	65.0	55.1	45.0	33.1	24.0
Boiling Point—Grav. Const.	168.5	173.0	176.6	180.0	189.0	194.0	171.0
UOP Characterization	12.5	12.19	12.16	11.9	11.62	11.2	12.22
Viscosity-Gravity	0.785	0.809	0.815	0.828	0.850	0.873	0.788
Aniline Point	184.0	169.5	185.0	163.6	141.0	127.0	188.0
Diesel Index	88.5	72.5	69.2	61.5	46.8	36.9	95.0



■ Fig. 4 – Relative engine roughness versus cetane number

such as boiling point, viscosity, gravity, or aniline point, all of which are interrelated with ignition quality. While some of these indices possess merit for certain types of fuel they are unsatisfactory when used for accurately indicating ignition quality of all fuels. Some of the various laboratory indices of ignition quality are:

1. Diesel index
2. Boiling point-gravity constant
3. Characterization factor
4. Viscosity-gravity constant
5. Aniline point

A detailed description of these various indices is to be found in the paper presented by Hubner and Murphy before the American Chemical Society, October 31, 1935, under the title "Effect of Crude Source in Diesel Fuel Quality" (see Reference 4 in Bibliography concluding this paper).

Table 3 gives index ratings calculated from the laboratory values of the several physical and chemical tests along with ratings obtained by the CFR delay method. Each index places most of the fuels in the same order or quality in which they are placed by the engine ratings. The last fuel, which is a product of the alkylation process, is an exception as it rates 24 cetane on the engine and yet gives a high index rating.

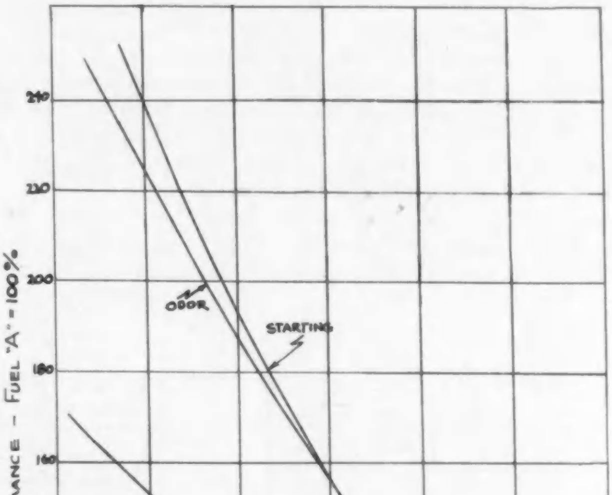
The recent work of the Full-Scale Group of the CFR Committee indicates that cetane number, as determined by the ASTM Procedure, gives a fair picture of how fuels perform in commercial engines relative to the effect of ignition quality.

■ Relation to Engine Roughness

Fig. 4 shows the relation between fuel cetane number and relative engine roughness. Five commercial engines were employed in developing this correlation. The cathode-ray oscillograph was used in obtaining the bulk of the engine roughness data. The pressure rise per degree of crank angle served as a criterion of the intensity of combustion knock.

Fig. 5 shows the relation between cetane number and such other important engine performance characteristics as smoke, combustion-chamber deposits, starting, and exhaust odor. The ignition-quality rating of diesel fuels, as it is a measure of the start of burning, gives a fairly good overall engine performance rating. With the cetane rating as a guide, the problem of the refiner is to produce a diesel fuel which will give satisfactory operation on all engines. The engine manufacturers, the engine users, and the armed forces are interested in such a fuel being available everywhere in the United States. Refinery processes have been developed to produce increased yields and improved quality gasoline. Those processes must be used also to produce the desired quality of diesel fuel.

Gas oils produced by straight fractionation of crude are



■ Fig. 5 – Relative engine performance versus cetane number

Table 4 - Laboratory Inspection Data

Crude Source	E. Texas	California	Oklahoma	Illinois	W. Texas	W. Texas	S. Texas	Louisiana	Kansas
Gravity, deg API	33.6	26.6	35.3	33.2	36.6	35.2	27.2	36.9	32.9
Flash, (PM), F	235	238	220	146	154	180	172	255	220
Viscosity, SSU	43.0	51.0	41.0	46.0	42.0	38.0	39.0	52.0	44.0
Pour, F	26*	20	20	40	50	20	B-10	60	10
Carbon Residue, %	0.01	0.023	0.01	0.017	0.01	....	0.053	0.013	0.012
Sulfur, %	0.18	0.60	0.250	0.300	0.30	0.69	0.13	0.280	0.210
Aniline Point, F	169.0	141.6	181.1	166.0	175.0	....	124.4	197.6	162.0
ASTM Distillation, F									
I.B.P.	460	468	409	340	350	375	368	502	382
10%	546	535	510	414	497	458	473	569	535
20%	554	557	529	540	540	488	491	584	546
50%	572	607	565	604	597	526	523	630	578
90%	631	672	629	718	674	648	581	755	638
End Point	700	729	677	762	715	701	630	775	667
Cetane No. - Delay	57.0	41.0	56.0	52.0	63.5	48.0	32.0	63.0	59.0
Diesel Index	56.8	37.6	63.9	55.1	64.3	....	33.9	73.0	50.9

Note: * - 33% Dewaxed
** - 40% Dewaxed
*** - 10% Bottoms

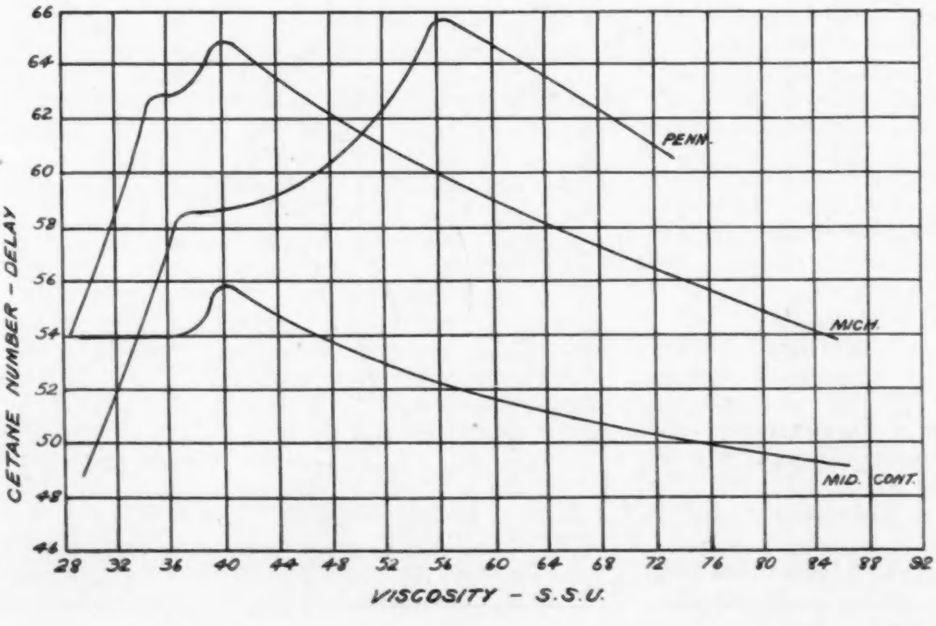
the chief source of diesel fuel stocks at present. This condition is primarily due to the high level of ignition quality which most of these stocks possess. Gas oils of this type are usually paraffinic in structure, particularly those stocks possessing ignition values above 50 cetane number, but they are likely to have pour points and viscosities too high to permit their direct use unless blended with lighter distillate or subjected to selective fractionation.

Table 4 gives several illustrative examples of typical straight-run gas oils obtained from a variety of crudes.

In order to obtain an idea of the amount of variation to be found in ignition quality of straight-run fractions from various crudes, a Michigan, a Mid-Continent, and a Pennsylvania crude were fractionated with steam in a laboratory dome still.

Cetane ratings were made on the different fractions from the kerosene cut to the heavier gas oils and residuals. These data are tabulated in Table 5.

The cetane-number variation of the different cuts from these three crudes is better illustrated by the curves in Fig. 6, wherein ignition value is plotted against the Saybolt viscosity of each fraction. Those distillates of around 30 or 32 viscosity have the lowest cetane number. In the Michigan and Mid-Continent crudes the highest ignition value is found in the fractionated material of about 40-sec viscosity but, in the Pennsylvania crude, this maximum is found in the material of 56-sec viscosity. It will be noted that, after the maximum cetane value is reached in each instance, the ignition quality drop-off is rapid for the higher viscosity fractions. As normal viscosity specification limits for a desirable diesel fuel range from 33 sec Saybolt to 43, it will be observed that this spread takes in a fair proportion of the available high-cetane-number material.



■ Fig. 6 - Variation of cetane number with viscosity of gas oil fractions taken from three crudes

on Various Straight-Run Gas Oils

Wyoming	Wyoming	New Mexico	Pennsylvania	Michigan	Indiana
36.0	33.7	29.3	37.4	32.9	33.6
196	190	190	240	184	184
45.0	44.0	46.0	46.0	39.0	46.0
60	45	B-10	40	25**	30*
0.015	0.009	0.015	0.024***	0.026	0.02
0.110	0.23	1.22	0.180	0.380	0.16
185.0	172.0	147.8	185.0	160.5	170.0
389	392	400	540	350	370
520	546	538	569	474	524
548	562	547	517	513	544
598	588	569	598	559	590
715	656	647	638	644	718
751	698	719	664	712	752
68.0	59.0	48.0	65.0	56.0	52.0
66.6	58.0	43.3	69.2	52.8	57.5

Commercial fuels are composed of complex mixtures of individual hydrocarbons, each having its own specific chemical, physical, and ignition characteristics. The properties of any commercial product therefore represent the sum total of the properties of these constituent hydrocarbons. In order to gain an idea of how the cetane value of different cuts of a straight-run diesel fuel varies, four fuels from a Pennsylvania, Mid-Continent, South Texas, and California crude respectively, were fractionated with steam in a laboratory dome still. These data are shown in Table 6.

Table 5 – Steam Distillation on Michigan, Mid-Continent, and Michigan Crudes

Cuts	Per Cent	Gravity	Viscosity at 100 F	Cetane No., Delay
Crude—Michigan—Gravity—41.6				
Naphtha	44.0	55.7	..	..
No. 1 – Kerosene	5.0	42.2	29	54
No. 2 – Gas Oil	5.0	40.0	35	63
No. 3 – Gas Oil	5.0	37.9	37	63
No. 4 – Gas Oil	5.0	35.1	40	65
No. 5 – Gas Oil	5.0	31.5	45	63
No. 6 – Gas Oil	5.0	27.3	57	60
No. 7 – Gas Oil	5.0	25.9	86	54
No. 8 – Gas Oil	5.0	23.0	..	..
Flux	15.0	....	..	..
Crude—Mid-Continent—Gravity—37.5				
Naphtha	44.0	52.7	..	..
No. 1 – Kerosene	5.0	38.6	30	54
No. 2 – Gas Oil	5.0	36.0	37	54
No. 3 – Gas Oil	5.0	33.9	40	56
No. 4 – Gas Oil	5.0	32.1	47	54
No. 5 – Gas Oil	5.0	30.0	59	52
No. 6 – Gas Oil	5.0	27.2	85	49
No. 7 – Gas Oil	5.0	25.5	150	46
No. 8 – Gas Oil	5.0	24.0	..	..
Flux	15.0	14.9	..	..
Crude—Pennsylvania—Gravity—41.3				
Naphtha	44.0	53.7	..	..
No. 1 – Kerosene	5.0	41.0	29	49
No. 2 – Gas Oil	5.0	39.1	37	59
No. 3 – Gas Oil	5.0	37.7	40	59
No. 4 – Gas Oil	5.0	35.9	45	60
No. 5 – Gas Oil	5.0	33.9	56	66
No. 6 – Gas Oil	5.0	32.0	70	62
No. 7 – Gas Oil	5.0	30.4	119	62
No. 8 – Gas Oil	5.0	28.6	213	54
Flux	15.0	....	..	..

Figs. 7, 8, 9, and 10 show the plots of cetane number versus Saybolt viscosity of the fractions from each of these diesel fuels. The range of pour points of these fractions is also indicated. From these data it would appear that fuel

Table 6 – Fractionation of Diesel Fuels from Various Crude Sources—Laboratory Inspection Data

Diesel Fuels from Pennsylvania Crude					
Gravity (API), deg	40.7	38.1	36.4	34.9	
Flash (P M), F	124	205	305	345	
Viscosity, 100 F (SSU), sec	36	42	48	56	
Kinematic Viscosity, 100 F (Cent.)	3.07	4.92	6.77	9.02	
Pour	10	35	45	55	
Carbon Residue, %	0.018	0.019	0.014	0.056	
Aniline Point, F	173.6	184.0	190.0	194.0	
ASTM Distillation, F					
Initial Boiling Point	253	390	566	600	
10%	444	540	594	633	
20%	469	559	601	637	
50%	539	584	620	649	
90%	586	626	658	673	
End Point	620	655	682	696	
Cetane Number	60.5	62.3	67.7	68.1	
Diesel Index	70.6	70.1	69.1	67.7	
Diesel Fuel from Mid-Continent Crude					
Gravity (API), deg	41.5	36.6	35.5	34.2	32.7
Flash (P M), F	136	215	240	280	315
Viscosity, 100F (SSU), sec	32	38	40	45	51
Kinematic Viscosity, 100 F (Cent.)	1.82	3.68	4.30	5.84	7.70
Pour	B-10	10	15	30	40
Carbon Residue, %	0.017	0.049	0.035	0.016	0.076
Aniline Point, F	151.4	165.2	168.6	173.8	176.6
ASTM Distillation, F					
Initial Boiling Point	328	433	470	523	563
10%	402	476	511	551	590
20%	416	490	523	563	597
50%	445	539	562	589	615
90%	515	626	633	647	662
End Point	576	679	678	691	697
Cetane Number	50.9	53.1	54.4	55.2	57.6
Diesel Index	62.8	60.5	56.3	59.5	57.8
Diesel Fuels from South Texas Crude					
Gravity (API), deg	32.1	28.0	26.2	25.2	
Flash (P M), F	140	215	255	285	
Viscosity, 100F (SSU), sec	33.0	39	44	50	
Kinematic Viscosity, 100 F (Cent.)	2.13	3.99	5.30	7.39	
Pour	B-10	B-10	B-10	B-10	
Carbon Residue, %	0.017	0.027	0.028	0.062	
Aniline Point, F	124.4	130.4	133.2	137.4	
ASTM Distillation, F					
Initial Boiling Point	334	444	498	526	
10%	406	474	523	552	
20%	420	487	528	566	
50%	455	517	548	574	
90%	511	578	607	625	
End Point	560	650	670	682	
Cetane Number	30.6	33.6	34.3	36.8	
Diesel Index	39.9	36.5	34.9	34.6	
Diesel Fuels from California Crude					
Gravity (API), deg	40.2	31.5	29.2	26.2	23.4
Flash (P M), F	148	190	220	275	350
Viscosity, 100F (SSU), sec	31	38	44	57	91
Kinematic Viscosity, 100 F (Cent.)	1.50	3.68	5.53	9.46	18.2
Pour	B-10	5	15	25	35
Carbon Residue, %	0.009	0.058	0.060	0.046	0.14
Aniline Point, F	135.6	139.8	141.6	144.4	145.2
ASTM Distillation, F					
Initial Boiling Point	352	395	424	506	602
10%	385	437	470	558	627
20%	393	456	495	576	635
50%	411	532	580	620	651
90%	464	654	663	678	698
End Point	576	713	725	700 (95%)	
Cetane Number	40.4	43.9	43.5	43.5	43.5
Diesel Index	54.5	44.0	41.3	37.9	34.0

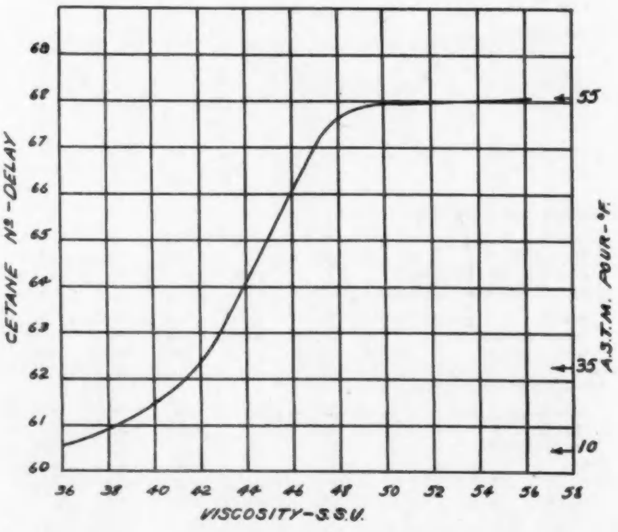
Table 7 – Laboratory Inspection Data on Various Cracked Distillates

Crude Source	Pennsylvania	Mid-Continent	Illinois	Calif.	Rocky Mt.
Gravity (API), deg	34.2	24.8	33.7	30.6	33.4
Viscosity at 100 F (SSU), sec	32	34	34	35.2	33
Pour	B-10	B-10	B-10	Below Zero	....
ASTM Distillation, F					
Initial Boiling Point	357	316	335	412	412
10%	450	446	429	438	440
20%	494	459	442	444	447
50%	593	487	468	471	462
90%	674	539	549	582	504
End Point	707	595	626	655	567
Cetane Number	34.0	29.7	37.0	34.0	40.0

viscosity or rather volatility, which is interrelated with viscosity, bears a definite relation to the ignition quality of fuels from a given gas oil. However, this relation does not hold true when applied to diesel fuels in general because of the influence of crude source, methods of manufacture, and so on. Although the higher viscosity fractions, in most cases, possess the better ignition quality, high pour points as well as restricted viscosity specifications may preclude their use.

Table 8 – Ignition Characteristics of Blends of Cracked Distillates and Straight-Run Stocks from Various Sources

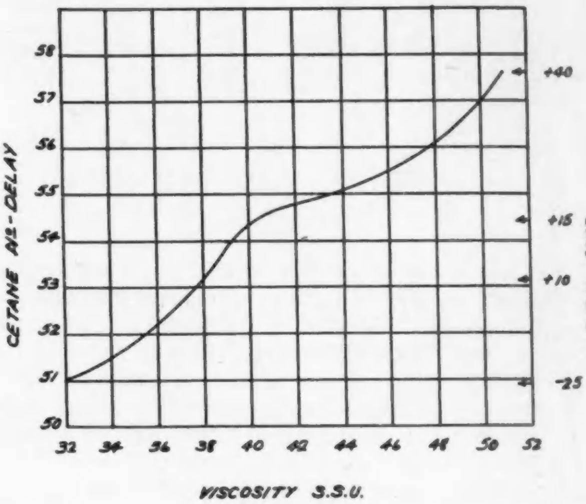
	Satisfactory Commercial Diesel Fuels				Unsatisfactory Fuel Blends	
Fuel No.	1	2	3	4	5	6
% Cracked Stock	65	61	50	25	50	50
% Straight-Run Stock	35	39	50	75	50	50
Cetane Number of:						
Cracked Stock	42	42	37	30	30	26
Straight-Run Stock	49	56	55	53	53	53
Cetane Number of Finished Fuel	45.2	48.0	46.0	45.9	40.0	38.0



■ Fig. 7 – Cetane-number variation with viscosity of Pennsylvania gas oil fractions

Cracked distillates, obtained as side-cut material in the cracking process, for the most part, are too low in ignition quality to permit their direct use as a diesel fuel. These fuels will usually range from a low of 30 cetane number to a high of around 47 cetane number. This depends, of course, on the nature of the charging stock, operating procedure, and degree of cracking. Table 7 gives laboratory inspection data on various cracked distillates.

Cracked distillates obtained by mild cracking, such as a No. 1 recycle stock, may be blended with suitable straight-run material to meet minimum fuel-ignition quality specifications of 45 cetane number. This is illustrated in Table 8. Blends of the more severely cracked distillates and straight-run fractions having resulting cetane values below 45 cetane number are generally considered unsatisfactory for use in many of the commercial engines of present design. Several examples of this are shown in Table 8.

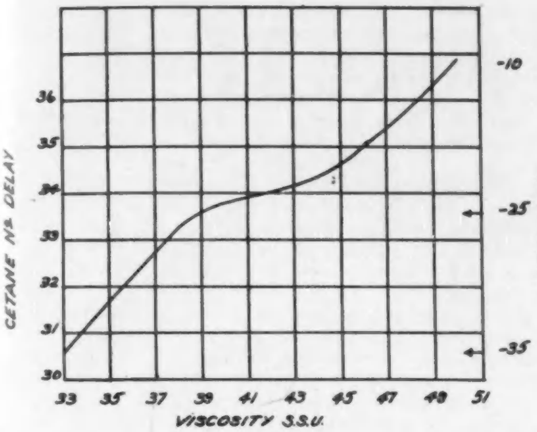


■ Fig. 8 – Cetane-number variation with viscosity of Midcontinent gas oil fractions

The use of ignition promoters offers interesting future possibilities for the economic utilization of a wider range of fuel stocks as well as for meeting special diesel fuel specifications. Quite a large number of these fuel dopes are available, although the more effective ones belong to the class of nitrates and peroxides. Thus we have acetone,

Table 9 – Data on Cracked Fuels Treated with Amyl Nitrate

% Amyl Nitrate by Volume	Cetane Number: Mid-Continent	Pennsylvania	Illinois
None	32.0	34.5	26.0
¼	37.0	40.5	....
½	40.0	44.5	32.0
1.0	42.0	50.5	36.0
2.0	49.0	57.5	41.0
3.0	....	....	46.0
Pour	−32	−25	−30
Viscosity, 100F, sec	32	31	33



■ Fig. 9 – Cetane-number variation with viscosity for South Texas gas oil fractions

peroxide, ethyl nitrate, amyl nitrate, and so on. Up to now, the absence of an overwhelming diesel fuel demand has been an important factor in discouraging the use of such materials. National defense requirements may alter the situation to some extent.

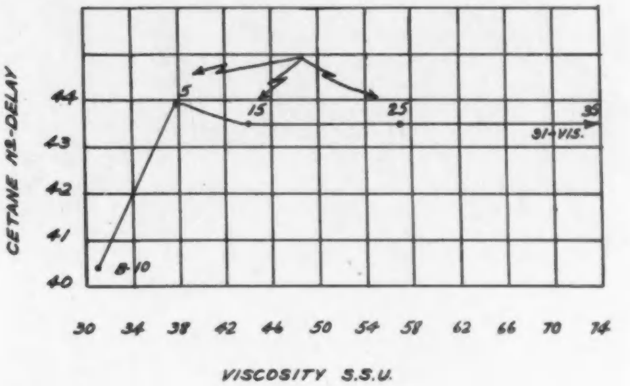
Table 9 and Fig. 11 give ignition and other laboratory data on several cracked fuels that have been treated by the addition of amyl nitrate.

In summarizing, we can conclude that the refiner has problems beyond his control which limit his ability to meet, economically, a wide variety of diesel fuel specifications. If the diesel-engine user is to continue to enjoy low-cost fuel, and if the difficulties traceable to the wide variation in the fuels now supplied by the different refiners are to be eliminated, then we must ask for a fuel which will satisfy the refiner as well as the consumer and which will meet the requirements of the commercial diesel engines now in use.

Table 10 – ASTM Diesel Fuel Classification (Grade 2-D) (Report of ASTM D-2 Committee for 1941)

Flash, min., F	140
Pour, max., F	20[a]
B, S and W, max., %	0.05
Carbon Residue (10% Btms) max., %	0.20
Ash, max., %	0.01
ASTM Distillation, F:	
90% Point, max.	650
End Point, max.	700
Viscosity:	
Kinematic (Centistokes)	2.0 min., to 6.0 max.
Saybolt (SSU), sec	32.6 min., to 45.5 max.
Sulfur, max., %	1.0
Corrosion	Pass
Alkali and Mineral Acid	None
Cetane Number, min.	45.0

[a] Lower pour points may be specified whenever required by local temperature conditions to facilitate storage and use. When operations are anticipated at atmospheric temperatures below an average daily minimum of +10F, lowering the pour point may also require a lowering of the minimum viscosity limit.



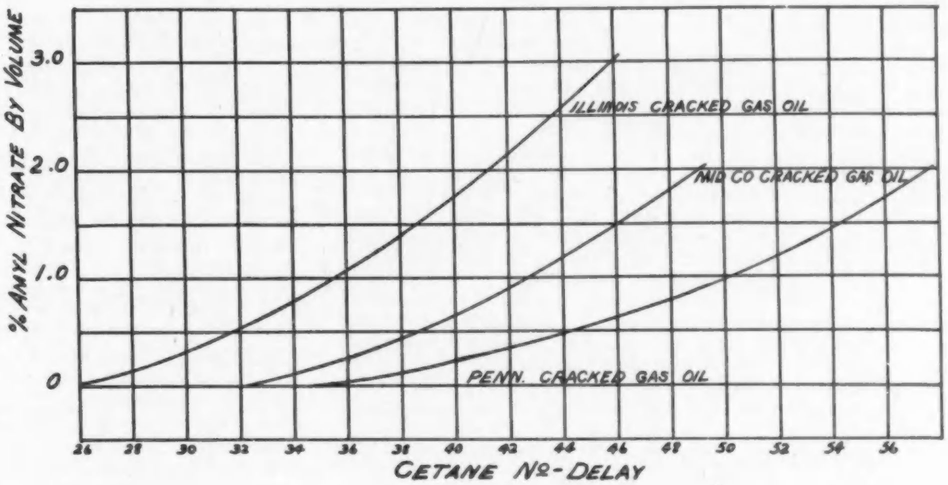
■ Fig. 10 – Cetane-number variation with viscosity for California gas oil fractions

It is with this thought in mind that Diesel Fuel 2-D of the ASTM Classification of Diesel Fuels (Report of Committee D-2 – 1940), is recommended for a universal diesel fuel suitable for mobile and high-speed equipment.

The specifications for this fuel are given in Table 10.

Several of the leading engine manufacturers have agreed that a fuel meeting this specification will give satisfactory engine performance and, with some tolerance allowed for cetane number and viscosity when fuel is required for low-

■ Fig. 11 – Effect of amyl nitrate fuel dope on ignition quality



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temperature operation, the refiners should be able to make such a fuel universally available.

Bibliography

1. "Evaluation of Diesel Fuels in Full-Scale Engines," presented by W. G. Ainsley, Report of Cooperative Fuel Research Committee, SAE Transactions, October, 1941, pp. 448-460.

2. "An Instrument for Measuring the Ignition Quality of Diesel Fuels," by W. H. Browne, SAE Transactions, April, 1941, pp. 148-153.

3. "The Development of Diesel Fuel Testing," by T. B. Hetzel, Bulletin No. 45, The Pennsylvania State College, 1936.

4. "Effect of Crude Source on Diesel Fuel Quality," by W. H. Hubner and G. B. Murphy, Universal Oil Products Co., Bulletin No. 176, 1936.

5. "Test Procedures and General Information in Current Use in the Development and Utilization of Aviation, Motor, and Automotive Diesel Fuels," compiled by the Cooperative Fuel Research Committee, May, 1941.

6. "Proposed Method of Test for Ignition Quality of Diesel Fuels," SAE Transactions, November, 1938, pp. 453-454, 464, 484.

DISCUSSION

Sets 50 as Top Practical Cetane Limit

—G. H. Cloud

Standard Oil Development Co.

THE authors are to be commended for their presentation of data that have helped to clarify the relation between cetane number and engine performance and between cetane number and other common fuel properties. They have made a worth-while contribution to the growing fund of knowledge of diesel-fuel performance that is overcoming misunderstandings that have heretofore existed.

With the problem of selecting suitable fuels for automotive diesel engines on the way toward solution, there remain only two other major handicaps to their general acceptance. One of these is the high cost of fuel-injection equipment, which is a manufacturer's problem. The other is cold starting, which is a problem for both the engine manufacturer and the fuel supplier.

Referring more specifically to the present paper, the CFR delay method for determining cetane numbers now being used is not an entirely satisfactory measure of ignition quality. It appears that the instrumentation for measuring delay has been developed to where it is acceptable, but the irregular operation of the CFR diesel engine limits the reproducibility of cetane ratings. Our laboratory has operated the "cetane valve" on the single-cylinder Caterpillar engine and has found the combination to be the best cetane-rating apparatus yet tried from the standpoint of reproducibility, speed, and ease of manipulation. It is estimated that the speed of rating with the cetane valve is about three times that with the present CFR ignition-delay apparatus.

Data obtained in our laboratory are in general agreement with the results presented in Fig. 5 showing the relation between cetane number and smoke, deposits, starting, and so on. All the data also agree in indicating that, except for cold starting, there is little to be gained by going above about 50 cetane number.

Laboratory tests and field observations in the U.S.A. and abroad have shown that the majority of popular types of automotive diesel engines will not start below about 25 F on conventional fuels without the use of cumbersome auxiliary equipment such as electric heaters and oversized batteries, small gasoline engines, and so on. If it were technically possible to solve the starting problem by setting an abnormally high cetane specification on conventional automotive diesel fuels, such a solution would be uneconomical. A better solution would be the use of special fuels for cold starting only.

The viscosities of the gas oils shown in Table 4 are higher than those generally sold as first-grade automotive diesel fuels on the East Coast, the average viscosity for this area being 34-35 SSU at 100 F. Reduction of viscosity of the gas oils shown to that prevailing in the East would result generally in a reduction in cetane number.

The relation between viscosity and cetane number for gas oils from various crudes is interesting. Our data have shown a similar relation, and have indicated that the optimum cetane number occurs at about 600 F in the gas oil boiling range. The results are in agreement with work by Wiezevich, Whiteley, and Turner[a] which has shown the minimum spontaneous ignition temperature of cuts from a series of gas oils to be at about 600 F boiling point.

Fig. 4 of a previous paper[b] shows the relation which has been obtained for the major diesel fuel characteristics affecting engine performance. You will note here that gravity increases directly with cetane number and that the 50% boiling point of the ASTM distillation increases directly with viscosity. For fuels of similar boiling range, the lower cetane stocks have the lower viscosities.

Referring back to the methods of estimating ignition quality, this chart has been quite helpful in estimating cetane number. Predicted cetane number for 34 of the 38 fuels, whose inspections are given in the present paper, falls within the range of zero to five units of the actual value. The inspections given represent an abnormal variety of stocks. In comparisons based on a recent survey covering 59 samples of automotive diesel fuels sold in the East Coast marketing area, 92% of the cetane numbers estimated from the chart fell within two units of the engine-determined values. On the basis of the relative importance of cetane and octane units, this is equivalent to predicting octane numbers to $\pm\frac{1}{2}$ point, which is considered rather good.

The relations between cetane number and viscosity shown in Table 5 and Fig. 6 for the Midcontinent gas oil are not in entire agreement with the results in Table 6, which show cetane number increasing with viscosity up to 51-sec viscosity.

The results on the effectiveness of the ignition promoter, amyl nitrate, are similar to those reported by the Universal Oil Products laboratories and observed in our own except that the increase in cetane number of the cracked Pennsylvania gas oil is much greater than would have been predicted from our experience. The high cost of the known promoters or their ineffectiveness has militated against their acceptance commercially. For example, the amyl nitrate is only about one-tenth as effective in increasing cetane number as tetraethyl lead is in increasing octane number. The fact that little is to be gained by increasing cetane number above about 50, which can generally be met satisfactorily by selection of gas oil stocks, tends to make the use of ignition promoters unnecessary.

In comparison with the gasoline antiknock situation, the diesel fuels ignition-quality picture looks bright for the following reasons:

1. Improvement in gasoline-engine design and performance have been accompanied by demands for higher and higher octane-number gasolines, whereas improvement in diesel-engine design appears to be in the direction of lower cetane-number requirements.

2. In gasolines, the higher the octane number, the more effective each octane unit becomes in permitting improved engine performance. With diesel fuels, the higher the cetane number, the less effect each cetane unit has.

3. Cetane number above that necessary for satisfactory operation from the standpoint of roughness, smoke, fuel consumption, is actually a handicap to a fuel because, for a given viscosity, an increase in cetane number is accompanied by a decrease in fuel heating value in terms of Btu per gal.

4. Blending values do not plague the diesel fuel manufacturer because cetane numbers of common gas oils blend according to a straight-line relationship.

5. There is no abrupt demarcation between satisfactory and unsatisfactory cetane values as there is for octane values. Roughness, smoking, and so on, increase gradually with decreasing cetane number.

6. There is no indication of the necessity for "road" cetane numbers. Nor is there any need for two or three types of cetane numbers

[a] See *Industrial and Engineering Chemistry*, Vol. 27, February, 1935, pp. 152-155: "Spontaneous Ignition of Petroleum Fractions," by P. J. Wiezevich, J. M. Whiteley, and L. B. Turner.

[b] See SAE Transactions, February, 1940, Fig. 4, p. 52: "Characteristics of Diesel Fuels Influencing Power and Economy," by A. J. Blackwood and G. H. Cloud.

ENGINEERING LIAISON and PRODUCTION CONTROL

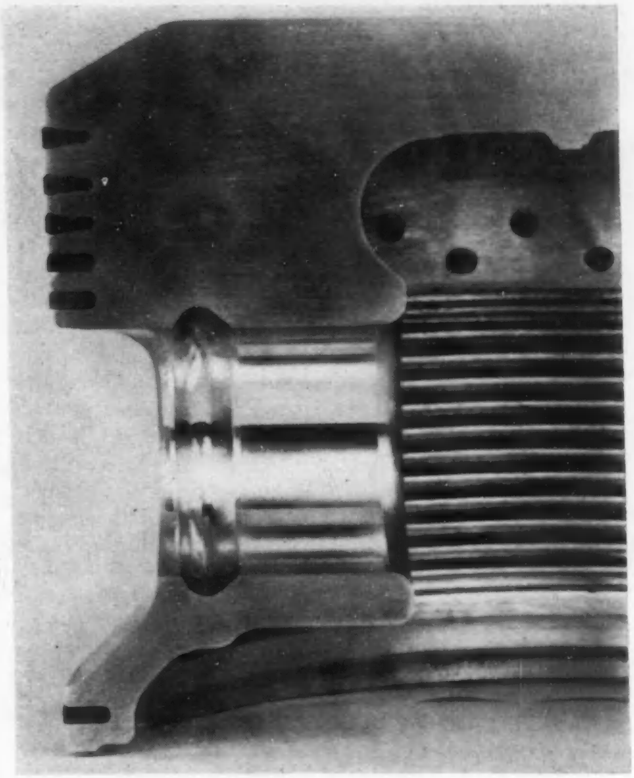
by DONALD U. KUDLICH
Wright Aeronautical Corp.

MANY of the differences between engineering and production units arise from the fact that the former are too far from the product which they have conceived to appreciate its deficiencies, and the latter are not qualified by experience to improve upon it. As a result, changes intended to correct a defect have bred others which, in turn, have necessitated additional modifications.

By the assignment of a third group, an engineering liaison, which is familiar with design requirements to observe the assembly and test, and other production phases of aircraft engines, the correction of basic faults and the elimination of unnecessary changes in design or procedures, have been possible. Examples of typical troubles and other methods of correction are cited in this paper.

★

THE AUTHOR: D. U. KUDLICH (M '35) engineering manager, Cincinnati Division, Wright Aeronautical Corp., joined the company in 1934 as an engine tester in the production test department. He has been associated with the Cyclone-14 engine since the first engine was built in 1936, and has been production liaison engineer on all models, including this engine.



■ Fig. 1 – Section of Cyclone 14 piston

THAT gulf which is reputed to exist between engineering and production units has frequently been romanticized by stories in which either an engineering genius or a production wizard casually solves the other's problems while surrounded by insurmountable troubles of his own. Unfortunately, industry cannot avail itself of these fictional heroes and must, therefore, arrive at the same end by commonplace methods.

To this end, there are two courses: first, to eliminate the gulf and make the engineering and production units think and act as one; or, second, to admit the fundamental difference and bridge the gulf.

By virtue of training and experience, the objectives of engineering and production personnel are widely separated. The engineer is concerned with the design, performance, and quality of the product and is reluctant to compromise on any points affecting these qualities. The production group is concerned with the fabrication of quantities of material at the lowest possible cost, and is inclined to view with disfavor changes which impede or upset their well-ordered processes. The elimination or even subduing of these fundamental differences is both unlikely and unhealthful, as the best productive efforts of either group cannot be secured by compromise.

The problem of bridging the gulf is one of satisfying both engineering and production units. In order to stimulate cooperation, it is necessary to determine and remove the sources of irritation. Interrogation of both groups at the main plant of the Wright Aeronautical Corp. at Paterson, N. J., revealed two major points of disagreement. Engineering personnel complained that a large number of unnecessary changes resulted from the inability of the Production Department to investigate variations in production engines, and that frequently deviations were made from recommended practices. The cry of the production

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Oct. 30, 1941.]

group was that there were too many changes in design, and many unreasonable design and process requirements.

The elimination of unnecessary changes and prevention of undesirable practices can obviously be circumvented by investigating every complaint or trouble to locate the basic fault, and by "selling" the need for adherence to recommended procedures if trouble is to be minimized. Much discussion has centered about the elimination of changes in production engines, but the pressure for release of advanced models, and the incorporation of new developments in production types, has thus far prevented any real progress in this direction. There can be no doubt, however, that the change in design processes on a given model can be accelerated materially so that the majority of the changes are made during the early stages of production, at which time tooling and other production facilities are likely to be liquid. This result can be accomplished by an intensive refinement program on production engines. So far as unreasonable design and process requirements are concerned, it can be conceded that they exist and that they may be reduced in number by determining their real value.

Some measure of success in accomplishing these aims has already been enjoyed by the Wright Aeronautical Corp. by the establishment of an engineering liaison which utilizes the facilities of the Production Assembly and Test Departments, and data secured from operation of production engines as a basis for changes in design or processes. The Liaison Engineer is responsible to the Chief Engineer, and investigates and coordinates change requests and production difficulties with the Project Engineers of the models affected, the numerous research branches of the Engineering Department, and the Manufacturing Department. Modifications which involve basic changes in the engine are referred to the Project and Experimental Engineers for action, whereas all other types are investigated on production engines. *The value of this experimentation on production engines cannot be underestimated, as variables which are not present in one or two experimental models can be easily evaluated from large-scale production tests.*

■ Three Types of Faults

Experience has shown that the faults which are evident in production engines are generally seasonal, chronic, or isolated. In the category of seasonal faults, piston-ring scuffing has long been outstanding; in fact, the coming of each summer has been announced by an insistent ringing of the telephone and a frantic request to "please do something to fix those piston rings."

■ Seasonal Faults

Although the exact cause of piston-ring scuffing had never been determined because production test results were not coordinated, a number of theories and as many cures existed. The investigation by the liaison group was begun by an evaluation of these theories:

First, there was the belief that lubrication was inadequate, and that "oiliness" of the lubricant must be increased. Compounds such as carbon tetrachloride, when added to the lubricating oil of groups of engines, provided no relief.

Second, it was proposed that lubrication was insufficient and that an increased oil flow would be necessary. Here again, no material improvement resulted when power-section oil flows were increased by means of additional oil jets.

Third, elevated temperatures were conceded to be a contributing factor. Efforts to reduce cylinder temperatures by decreasing air leakage between cowls and baffles and by lowering oil temperatures to the minimum permitted by contract specifications, were only moderately effective.

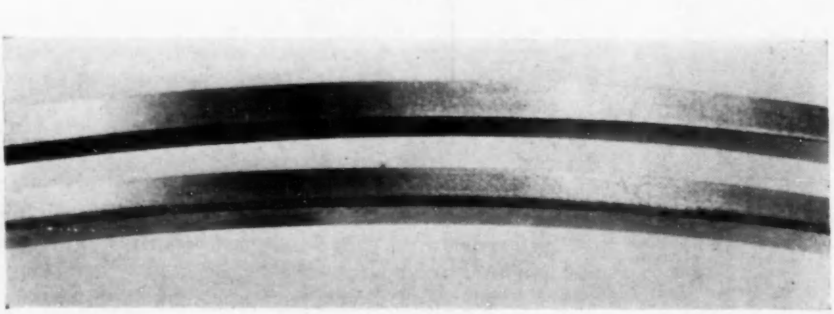
Fourth, ring butting was advanced as a cause. Tests with large gaps showed no improvement.

Fifth, the claim was made that the piston-ring arrangement was too severe, and that revisions would be necessary to increase the oil consumption and permit more oil to pass by the rings. Scuffing was almost entirely eliminated by this method.

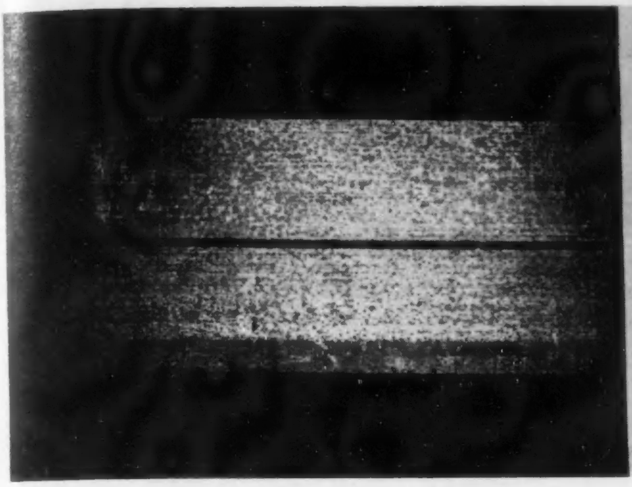
Sixth, material, and changes in the manufacturing process were suspected since repeated tests under identical operating conditions would eventually produce a satisfactory combination. It was found that material supplied by several vendors showed different degrees of scuffing resistance and uneven wear. The process of lapping rings in cylinder barrels as practiced on experimental engines was also conceded to increase scuffing resistance.

Of all these devices, the latter two, namely, modifying the piston-ring arrangement, and altering the manufacturing process, had sufficient merit to warrant further investigation.

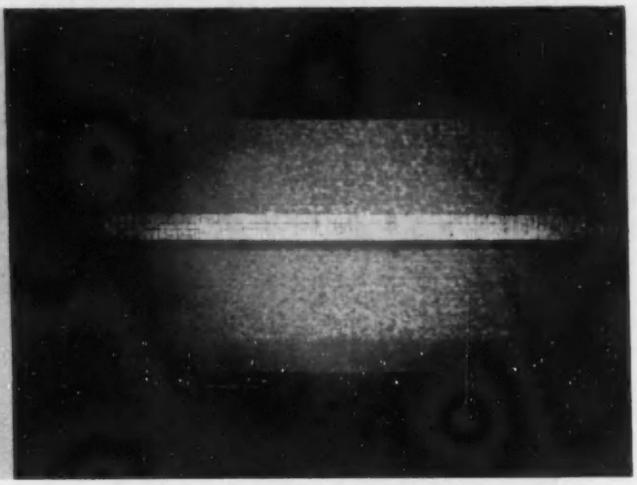
Fig. 1 is a sectional view showing the general arrangement of the "Uniflow" piston used in all single- and double-row Cyclones. Compression rings are of the non-sticking wedge type with a ½ to 1-deg tapered outer face. Oil-control rings, which are used in the two grooves just above the piston skirt, are flat-sided compression rings having a 1½ to 2-deg tapered outer face. Oil removed from the cylinder walls by the control rings is led to the inside of each piston through 12 holes in the lower and 10 holes in the upper ring groove. The holes are located in the thrust area of the skirt only, and are staggered as can be seen from the right side of the photograph. The oil pumper ring is likewise a flat-sided and tapered outside diameter ring, but is inverted so that contact with the cylinder occurs at the top side. Two modifications of this



■ Fig. 2 – Variation of circularity and lapping on new piston rings responsible for uneven wear



▪ Fig. 3 – Sections of new competitive rings showing complete absence of lap contact on upper ring – 10X diameters



▪ Fig. 4 – Sections of new competitive rings showing scored appearance of rings "lapped" with lubricant and satin finish obtained with abrasive lap – 10X diameters

arrangement were tested as a means for providing sufficient oil for lubrication.

1. The simple expedient of inverting various combinations of rings, produced some useful information. It was found that, contrary to general belief, compression rings on a piston of good design normally contributed little to oil control, and that anything short of their complete removal was possible, provided the oil-control rings were functioning properly. Oil-control rings were critical to the radius on the bottom or scraping edge and any radius in excess of 0.002 or 0.003 in. was conducive to high oil consumption. This radius could be varied for differences in oil viscosity as might be caused by summer and winter operation – the sharp edge being necessary for summer operation or low-viscosity oils. Combinations of inverted compression rings and rounded-edge control rings caused oil consumption of 60 lb/hr; *an increase of 500% over the normal average consumption of the double-row Cyclone 14 did, however, prevent ring scuffing.*

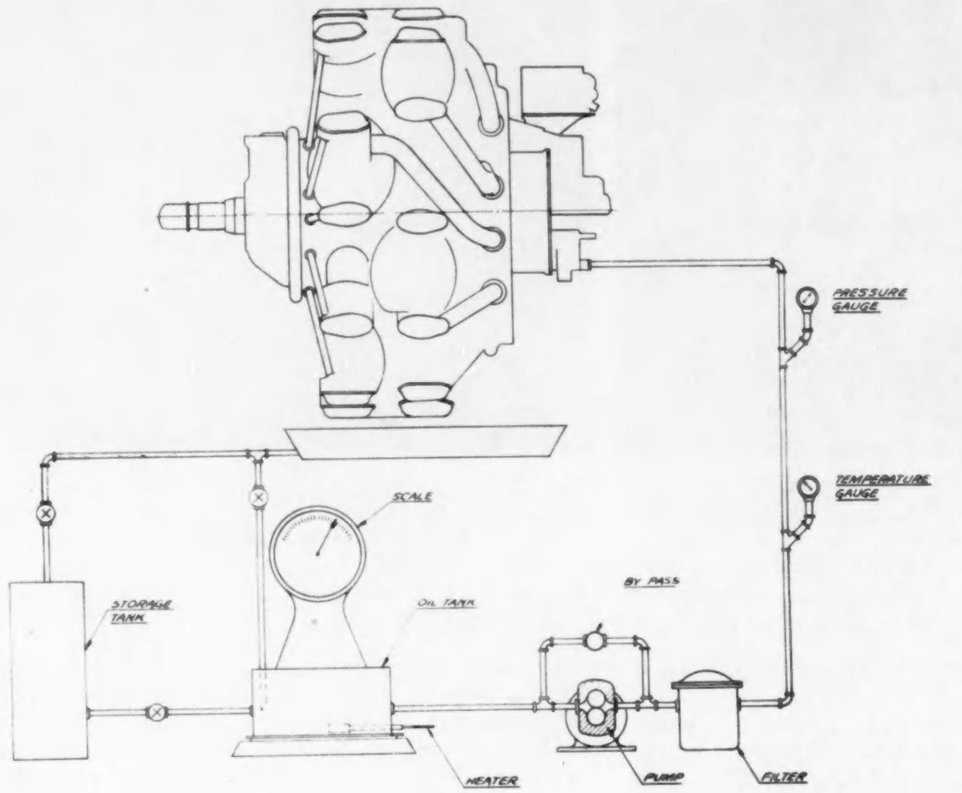
2. The practice of adding a radius or chamfer to the bottom edge of the oil-control rings, a cure of long standing, was stopped in view of the unpredictable effect on oil consumption, and the radii applied to the top and bottom of compression rings only. Moreover, the radii were added to the drawing and formed by the vendors in order to prevent the formation of feathered or sharp edges on hand-worked rings. This change reduced the number of scuffed rings by 50%.

The second device, the testing of different types of cylinder-barrel finishes, demonstrates the value of experimentation on production engines. Cylinders were finished by four different methods: (1) corkstone honing; (2) stone honing; (3) draw honing; and (4) lapping.

There was no difference in the finish of cork- and stone-honed cylinders. The abrasive impregnated soft cork hones were used only to follow the contour of choked bore cylinders, and the stone hones for straight bores. Cork stones were 100-grit spaced equally around the bore in four pairs of two stones, each pair being separated by a fiber wiper. They were operated at 74 rpm with 28 to 30 oscillations per min, using a compound of 90% kerosene and 10% cutting oil as a coolant and lubricant. A finish of approximately 2 to 6 micro-in. was obtained in 5 min by this method. Except for stones, all other finishes were produced in the same manner. Draw honing was done with interlocking 37500 vitrified stones which covered the entire cylinder surface and were reciprocated only during the stroke. The hone was rotated approximately ½ in. at the end of each stroke. Lapped barrels were first honed by means of 80-grit corkstones, after which they were lapped for 2½ min or 100 strokes, using an M600 compound and dummy piston rings. Several lots of these cylinders were prepared, and the condition of both rings and cylinders carefully noted after the production test. It was soon apparent that it would be extremely difficult to secure an honest classification of the parts, as a ring classified as good by one observer who had been looking at a large number of bad rings might be classified as bad by an observer who had just completed an inspection of a number of good rings. Fortunately, a sufficient amount of data had been accumulated by this time to show the way out, and the engines thereafter rated on the *number of scuffed rings only,* without regard for the degree. The data from the first lot of 21 engines are shown in the following table:

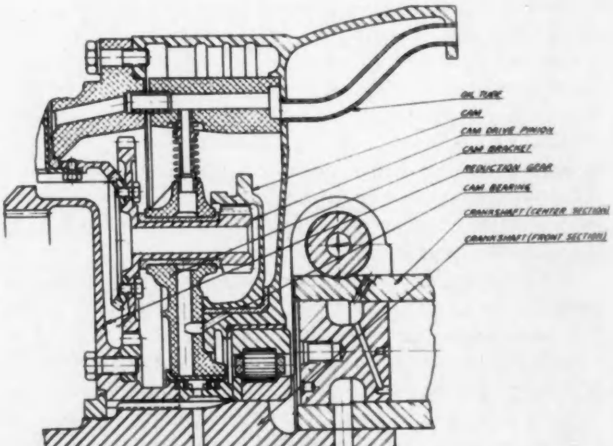
Type of Finish	No. of Engines Tested	No. of Engines with Scuffed Rings
1. Cork-honed	4	1
2. Stone-honed (conventional production)	8	6
3. Draw-honed	1	1
4. Lapped	8	0

No differentiation is made for the various degrees of finish as there was little difference noted over the range tested. It is obvious from the foregoing table that the production method of finishing was the least effective, and that lapping was the most effective. In analyzing the reason for the outstanding performance of the lapped barrels, it was noted that piston wear was greater with lapped cylinders than with honed cylinders of equal roughness. It was apparent, therefore, that some lapping com-



■ Fig. 5 – Apparatus for flow testing

■ Fig. 6 (below) – Cam drive mechanism – double-row Cyclone 14



pound was retained by the cylinder despite elaborate washing precautions. As Experimental Department practice had shown it to be unnecessary to re-lap cylinders after each test unless rings were changed, the major, but heretofore unnoticed, change must be on the rings in the form of rounded edges and altered surfaces. A review of the results of tests of rounded-edged rings and lapped barrels, and inspection of new rings confirmed this conclusion.

Fig. 2, of two new competitive rings, shows the non-uniformity responsible for uneven ring wear. Both these rings have lapped, tapered faces, the upper ring having been "lapped" with a kerosene and oil mixture to a finish of approximately 5 micro-in., and the lower ring to approximately 24 micro-in. Fig. 3 shows sections of the rings at 10 diameters, and the complete absence of contact with the kerosene lap on a section of the upper ring. Fig. 4 shows the scored appearance of the kerosene and oil-lapped section.

When run in cylinders having a finish of 2 to 6 micro-in., the abrasive-lapped ring with the addition of an 0.015-in. radius was equal to, or better in scuffing resistance than those produced by lapped barrels, and required no modification to the lubricating system.

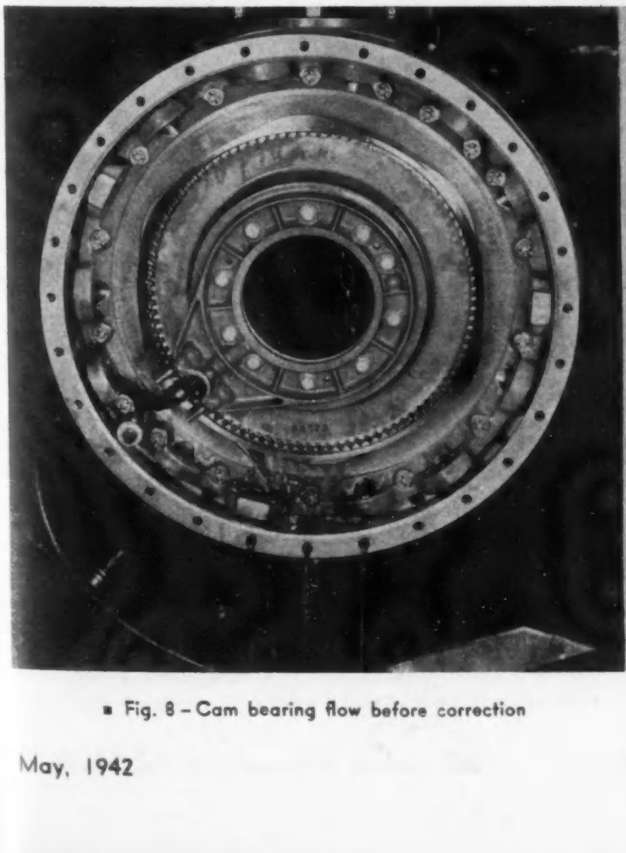
From the foregoing tests we may conclude that the least wear and greatest scuffing resistance is obtained when one of two lubricated rubbing surfaces is smooth, and the other rough. This phenomenon is probably due to the fact that the rough surfaces are supported by innumerable peaks surrounded by oil which prevents the mass of the bearing from reaching temperature values which would melt and tear the surface. As a result of this development, increased aircraft-engine production was possible during the past

■ Fig. 7 – Excessive bushing chamfer causing oil leakage

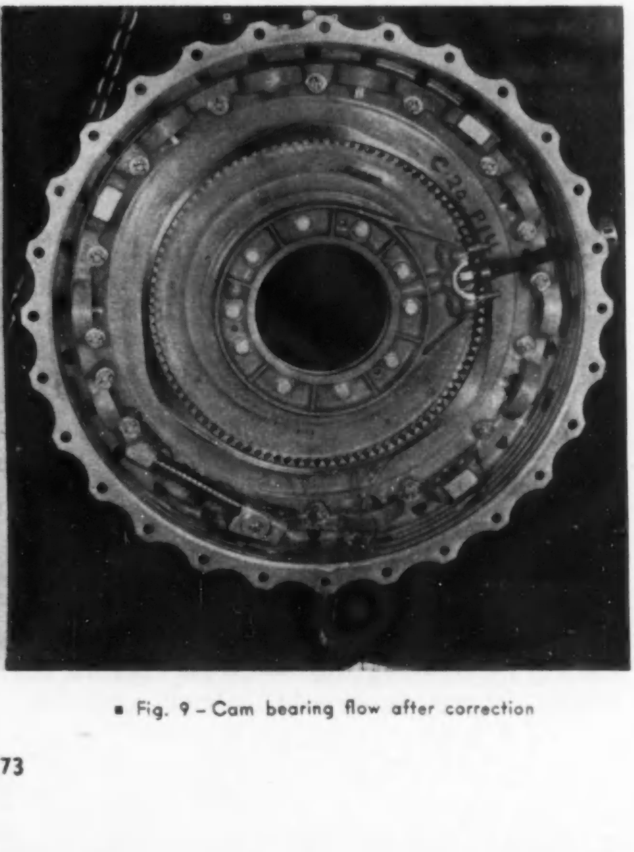
summer. However, this does not complete the liaison engineer's work. The performance of new models at higher ratings must be carefully watched for such trouble, and any recurrence forestalled by active development now.

■ Chronic Troubles

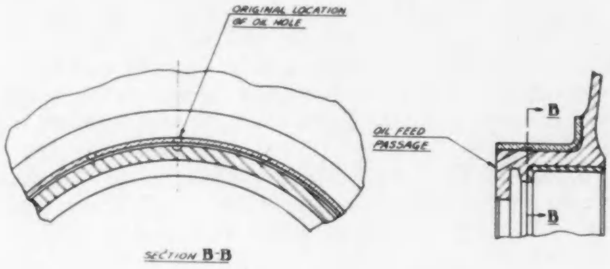
Many of our chronic troubles arise from the rigidity of aircraft-engine specifications, the necessity for adhering to their requirements, and the fact that the data therein are based upon the performance of a limited number of experimental models, and upon past experience. There are a number of parts the performance of which cannot be predicted from drawings or from tests of one or two units. The lubrication system is particularly vulnerable in this respect. The tolerances necessary for manufacturing, assembly, and proper functioning of the parts may vary the oil flow to such an extent that many production engines will exceed the flow limits. This factor would be unimportant if equipment such as oil coolers were selected to fit the engine installed in each airplane but, as it is not, engines having oil flows in excess of specified limits must be corrected. The identity of the parts or groups of parts causing flow variations can be best determined by observation of a number of production engines. A static flow rig which is shown in Fig. 5 is used for this purpose. Oil from a reservoir is pumped through the engine oil pump inlet to all parts of the engine. The leakage from bearings and other points is drained from the sump into a pan and then to a storage tank. Various components or sub-assemblies are tested in the same manner except that oil issuing from the smaller units is caught in small pans and weighed on a balance scale. It is essential that oil entirely free from any foreign matter be used to prevent clogging of closely fitted bearings. Oil having a viscosity at 100 F equal to that of lubricating oil at 220 F is used for all tests for ease in handling the parts and to prevent burning the operator if a plug is blown out or an oil line bursts. The pressure is regulated to 80 psi for tests of complete engines and the supercharger rear housing which contains all accessory drives. All sub-assemblies are tested



■ Fig. 8 – Cam bearing flow before correction



■ Fig. 9 – Cam bearing flow after correction



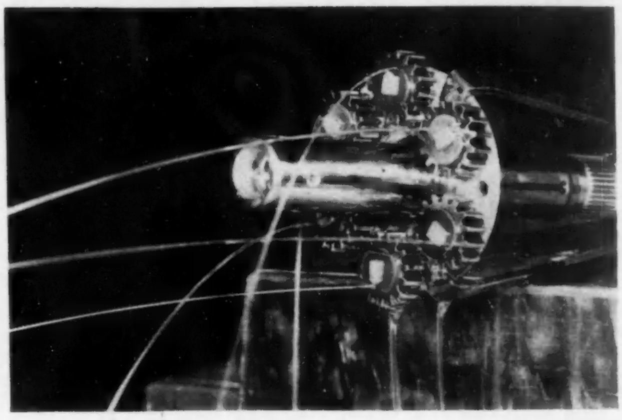
■ Fig. 10 – Assembly change to reduce oil flow to cam bearing

at 60 psi to simulate actual operating pressures. Under these conditions it has been found that the static flow of a complete engine is approximately one-half the kinetic flow at rated power and speed. This relationship is not true for the various sub-assemblies, as the pumping capacity of each bearing and journal varies with the shaft speed. Once the relationships for each assembly are established, however, it is an easy matter to predict results of full-scale engine tests from static tests. The major advantage of the rig lies in the fact that *points of leakage or flow which serve no useful purpose may be seen, and guess-work eliminated.*

Fig. 6 shows the general arrangement of the cam drive mechanism of the 14-cyl double-row Cyclone engine which is being built on a line production basis. The cam mechanism is lubricated from two sources. Oil is fed from an internal oil passage at the top of the engine to the top of the intermediate cam gear shaft bracket. Passages in the cam bracket distribute the oil to the intermediate cam gear shaft bushing and to the cam bearing oil feed. A supplementary supply of oil is fed from the crankshaft through a rotating piston-ring type oil seal to the top of the inside diameter of the bracket and thence to the cam and intermediate driveshaft bushing. What appeared to be an excessive amount of oil for lubrication of the intermediate cam drive gear shaft proved to be leakage upon examination of the parts. By reducing the large 0.03-in. chamfer shown in Fig. 7 of the intermediate cam drive gear shaft bushing, to 0.002 in. maximum breakedge, the static flow of each of the two cam drives was reduced 2 lb per min. Figs. 8 and 9 illustrate the reduction in flow from the cam bearing which was obtained merely by rotating the bearing on the crankcase so that the oil passage in the crankcase was not aligned with the oil holes in the cam bearing. This drawing change, which reduced flows of each of two cam bearings from 8 to 3 lb per min, is shown by Fig. 10. Figs. 11 and 12 show the reduction in flow from the two-speed supercharger intermediate pinion assembly which was obtained by refinement of tolerances. Fig. 11 has been aptly termed the "fireboat" version of the two-speed supercharger drive. Another type of change which facilitated manufacture and reduced flow is shown by Fig. 13. The method of manufacture shown by the lower half of the drawing requires a close concentricity tolerance between the starter coupling spline and pilot in the accessory drive shaft. With this design, 0.001 to 0.003-in. clearance was necessary between the accessory driveshaft and starter coupling. By substituting a tight-fitting solid plug shown in the upper half, the pilot and concentricity requirement on the starter were eliminated. Modifications of the type described have been effective in *preventing hasty, detrimental changes which production-minded persons might demand to prevent production stagnation.*

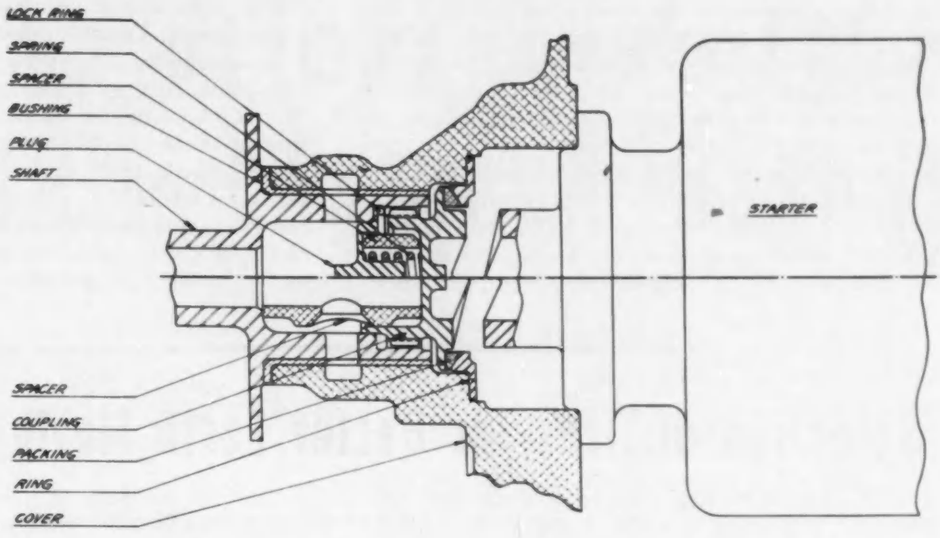
■ Isolated or Occasional Faults

The expression, "isolated failure," is probably responsible for much of the laxity prevalent in the prevention of additional failures of the same type. The pressure upon the aircraft-engine engineer for the improvement of parts which repeatedly cause production or service troubles is a contributory factor as, relatively, the isolated failure may seem unimportant. That a failure has occurred, however, is indicative of the possibility of recurrence, and with a frequency and severity to completely stop production. Moreover, the fact that the lives of persons using air



■ Fig. 11 – "Fireboat" two-speed supercharger drive before correction

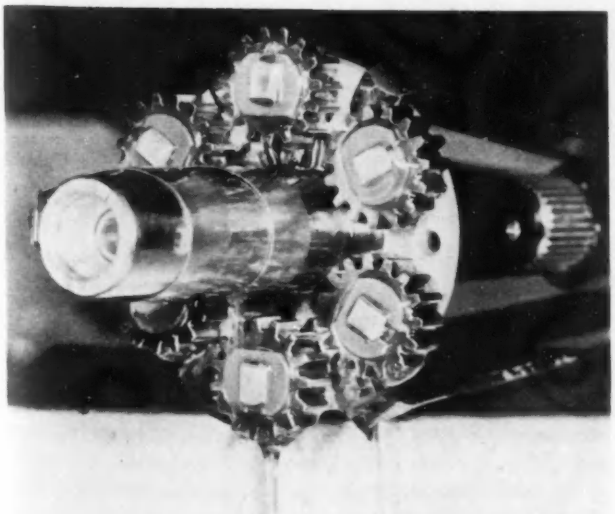
transportation is dependent upon the uninterrupted performance of the powerplant, makes it imperative that any defect be immediately corrected. In the prevention of troubles of the occasional or isolated type, the services of the liaison engineer may be utilized to best advantage for, by observation of engines at inspection after the production test, troubles may be found and corrected before progressing sufficiently to become serious. One typical case which might have impeded the production schedule was an interference between the intermediate cam gear pinion and cam which was possible with an accumulation of tolerances. Evidence of slight interference noted after production test initiated a re-inspection of parts to check conformity with drawings. As the parts were not deficient in this respect, dimensions and tolerances of the cam drive mechanisms were checked and a possible interference found. The process of eliminating such an interference appears rather simple upon a cursory inspection, but the necessity for maintaining interchangeability of parts, and the removal of a considerable quantity of metal to preclude any recurrence, required some time for "juggling" the figures. The cure was no sooner found and set in motion, than the fault became chronic! The need for a hasty and compromise correction was obviated, however, by the action



■ Fig. 13 – Design change to eliminate oil leakage

taken on the isolated case, and all parts repaired to the satisfaction of the production and engineering units with a minimum of delay. Past experience has shown that action is not effective unless engineering personnel are accessible and prepared to look at the evidence when it is available. The liaison engineering unit fulfills this need.

The maintenance of charts and graphs of the numerous engine functions has also proved invaluable in locating obscure reasons for sudden changes in performance. This condition is especially true when the lubricating system is affected, as there are so many bearings, journals, and seals of different types involved that the process of locating the faulty one would otherwise prove to be difficult and tedious. By means of a graph showing the oil flow and heat rejection of each engine, an increase in flow and heat rejection



■ Fig. 12 – Two-speed supercharger drive after correction

was noticed. A review of changes in processes or design which had been recently incorporated in engines did not in itself indicate any possibility of trouble but, by coordinating the graph and changes, a definite relationship was found between the introduction of the master rod plating process and the increased flow. Theoretically, there should have been no change as drawing dimensions affecting flow were unchanged. It was necessary, therefore, to investigate the entire process of master-rod manufacture and secure measurements of the bearing size before plating, after plating, and after the initial test. A perusal of these data showed the plating process to be at fault, bearing clearances being increased by one-third after production test. Plating processes were then investigated by engineers who developed a more durable plate requiring less plating time. Here again, active investigation resulted in the correction of a seemingly insignificant fact to the complete satisfaction of production and engineering personnel.

The liaison unit has a number of other useful functions. The dissemination of engineering information to the Production Assembly and Test Departments has expedited production at a time when we can ill afford delays arising from petty troubles. Typographical and other errors in engine specifications, assembly, and test specifications require interpretation and deviations until new specifications can be prepared. Immediate verification of the fault and the granting of deviations precludes delays from troubles of this nature.

The fact that production personnel are not familiar with new devices or the application of changes to existing models is also cause for dissatisfaction. By explaining the purpose and effect of changes and demonstrating their application, it has been found that production personnel frequently take advantage of the personal contact afforded by such a procedure to point out ways and means of improving the product.

■ Conclusion

To meet the ever-increasing production demands for national defense, a better understanding of the problems involving engineering and production groups and their

wholehearted cooperation are essential. We cannot expect these things if operations are based upon opinions rather than upon fact. In determining the correction for each trouble, regardless of the type, we can too easily become involved in personalities. As the refinements based on personal reactions have not always resulted favorably, the engine has been the victim. By observation of the variations in production engines, the liaison engineers have been able to *find out what the engine wants* so that the personality problem becomes non-existent. The story that the engine tells cannot be refuted. The engine has also indicated where short-cuts in manufacturing, inspection, assembly, and test procedures are possible.

The experience gained from observation of the liaison engineering unit has been instrumental in determining the type of organization required for operating purely production plants such as the new Cincinnati Division of the Wright Aeronautical Corp., which represents the first attempt of the aircraft industry at truly mass production. There is every reason to believe that the close cooperation developed among the various branches of the Engineering and Production Departments by determining the facts from masses of production data, will produce new records for quality, production, and costs.

Selection of Tires for Better Earth-Moving Efficiency

ALTHOUGH the moving of earth on large pneumatic tires has been done for only a few years, we have learned much in that time that can be passed on to the operator in the interests of economy and better performance.

We who have to counsel with designers on the application of tires to new designs of equipment, have tried to profit by these experiences, and we recommend only such tire equipment as we believe will give complete satisfaction to the user. By this we mean satisfactory overall performance of the machine as well as of the tires themselves.

Generally, there are five fundamentals to be decided upon when the user contemplates the purchase of new equipment. These are: 1. tire size; 2. tire type; 3. rim size; 4. inflation pressure; 5. single or duals.

Let us take these up one by one.

Tire Size – The tire size should be selected after thorough analysis of the operating conditions and can be determined only with full knowledge of: (1) empty weight of the equipment; (2) capacity of equipment; (3) weight of material to be hauled; (4) distribution of gross load; (5) speed; (6) length of haul; (7) condition of haul road.

With this information before him the user can then discuss his problem with a competent tire man and arrive at the proper tire size.

Tire Type – There are no basic construction differences aside from the tread patterns, each of which has its own field of application.

Tire types can be divided into three general groups: (1) maximum traction; (2) moderate traction; (3) free rolling.

Maximum-traction tires are best suited to soft going. The wide spacing of the traction bars provides the necessary cleaning quality under adverse conditions – a feature which is necessary in soils that clog or foul the design. It is quite apparent that continued spinning of the wheels when there is no forward motion will result in a rapid "digging in" and consequent complete stalling of the machine.

The moderate-traction tire overcomes many of the foregoing objections, although it should not be construed as eliminating the necessity for the maximum traction type. This less aggressive design can spin with less tendency to dig, which greatly improves the operation of the equipment in many cases. This design is more closed up, giving more protection against rock damage, also smoother operation on hard surface haul roads, and in general can be expected to give better all-round performance and longer life.

The difference between free-rolling tires and other types is entirely in the tread pattern which obviously is not well adapted to drive-wheel service. In soft soils the design does not clean satisfactorily, thus causing a loss of traction. In general the skid depth on these tires is less than those designed for traction wheels, resulting in lower mileage expectancy. Conversely, there is ample life and resistance to normal rough usage when applied to free-rolling wheels.

Rim Size – Equipment designers have cooperated very well in the matter of applying correct rims. Users should likewise recognize the importance of this part of the installation and not try to oversize without rim or wheel changes.

Inflation Pressure – We have previously pointed out the value of lower inflation pressures in reducing the tendency to rock-cut or fail by impact. Special consideration must be given this feature if equipment is being purchased for rocky operation. Lower pressures are possible when using larger-section tires.

There are operations where flotation of the equipment must be given most careful and serious consideration. In soft soils, flotation is gained only through the application of larger tires to operate at lower air pressures, giving increased ground contact area.

Singles versus Duals – Advantages and disadvantages can be set forth for both single and dual equipment.

Single installations have decided advantages in rough ground operation. Tires are not subjected to momentary extreme overloading as in the dual assembly when one tire carries the entire load. There is no space to pocket stones and trash such as between dual tires. Instances have been known where this packed material has badly bent the dual spacer band.

In soft going it is generally recognized that single tires pull easier, making it possible to operate in a higher gear and so increase the speed and overall efficiency of the equipment.

Duals are slightly cheaper – always a very attractive feature. Also, they lower the center of gravity and increase the lateral stability which is sometimes desirable in slope work.

Excerpts from the paper: "What the User Should Know About the Selection and Care of Tires for Better Earth-Moving Performance and Economy," by L. W. Fox, sales engineer, The Firestone Tire and Rubber Co., presented at the National Tractor Meeting of the Society, Milwaukee, Wis., Sept. 25, 1941.

Problems of Changing from ALUMINUM to CAST-IRON PISTONS

by WILLIAM S. JAMES

Chief Engineer, The Studebaker Corp.

THE principal problem in changing over to cast-iron pistons at this time is in designing them for low weight and adequate strength, Mr. James concludes. In the final analysis, he points out, the design of the cast-iron piston is controlled by the ability of the foundry to cast thin sections to close tolerances. As soon as this part of foundry technique improves, cast-iron pistons can be made lighter and, possibly, as light as the present designs of aluminum pistons, Mr. James states.

The greater weight of cast-iron pistons, imposing increased loads on the bearings, connecting rods, and piston pins, is much more difficult for the designer to overcome than are such problems as oil consumption, engine friction, piston scuffing, piston slip and rattle, and similar factors, the author avers.

Mr. James' paper is made up of a series of experiences of various engineering departments who have undertaken the switch from aluminum to cast-iron or cast-steel pistons. Compiled at the suggestion of the Passenger-Car Activity Committee, the paper gives comparative data covering all important phases of piston engineering.

THE AUTHOR: WILLIAM S. JAMES (M '18) Studebaker's chief engineer and SAE Councilor for 1942-43, joined the National Bureau of Standards while still working for his B.S. in mechanical engineering at George Washington University. Prior to joining Studebaker as research engineer in 1927, he had been assistant technologist with Associated Oil Co.

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AT the suggestion of the Passenger-Car Activity Committee of the Society, the engineering departments of several car manufacturers were asked if they would outline their experiences in substituting cast-iron for aluminum as piston material. The author was asked to correlate this material into a paper for presentation to the Society. Such correlation has been limited to a few general conclusions because of the extremely interesting manner in which the material was presented by the several engineering departments and because, as presented, it represented individual viewpoints and interesting independent solutions of a common problem. The material is presented as submitted except for some editing to eliminate commercial names and references to the sources of the material. The editor-author felt that any attempt to correlate the information submitted would result in an incorrect and confused picture of the varying and interdependent factors involved.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 15, 1942.]

In order to make these several experiences clearer, a sectioned perspective drawing of the final piston design as released is shown for five of the seven piston designs discussed.

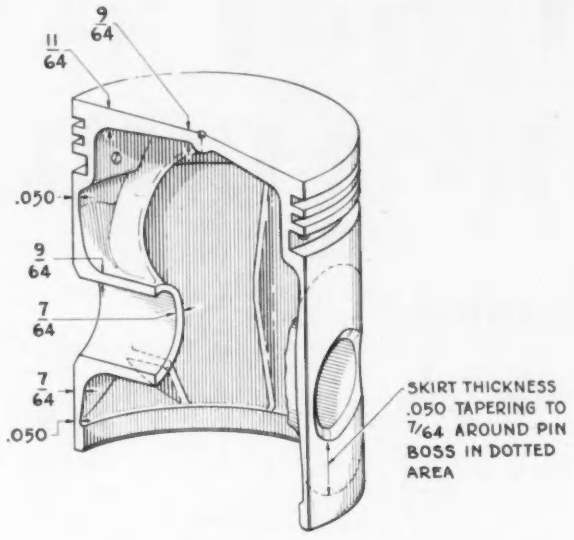
■ Piston Design "A" (See Fig. 1)

1. Connecting Rod

(a) We kept the same connecting-rod and wristpin construction, pin-locked-in-rod-type, in the engine in order to preserve interchangeability in the field.

(b) The material of the connecting rod was changed from water-quenched 1040 steel to an oil-quenched SAE X-1335 steel. This gave better physical properties and freedom from water cracks, and helped to produce a rod more able to stand the additional loads of the cast-iron piston.

(c) We changed from a construction, where babbitt was tinned direct to the big end of the rod, to boring the big end of the rod with a carboloy tool and using a steel-backed



■ Fig. 1 – Piston Design "A"

liner having a babbitt of 0.005-0.007 in. thickness. This liner, we find, has about twice the life of the babbitted rod, due largely to the fact that we cannot babbitt a rod and have the babbitt as thin as with a liner.

2. Lubrication

No fundamental changes were made in the lubrication of the engine.

3. Life Test

Our standard life test on the standard engine with aluminum pistons is 4400 rpm for 100 hr without breakdown. With iron pistons, we had to cut the speed to 4000 rpm and, at the end of 50 hr, the old babbitted rod showed distress. By using the rod liners mentioned in Item 1, we were able to go at 4000 rpm with the iron pistons for 100 hr.

We can summarize by saying that, by bettering the rods and liners, we have obtained about as much out of the iron pistons at 4000 rpm on the bearings, both main and rod, as we do with the aluminum pistons at 4400 rpm.

4. Main Bearings

These bearings were changed some time previously to the 0.005-0.007 in. thick babbitt bearings. No further change was made when we went to iron pistons. Main bearing life did not seem to be a serious factor.

5. Iron Pistons

(a) We specify a molybdenum content in the iron in order to give it higher physical properties – to get a tougher iron and still make it machinable.

(b) The ring combination in the piston is the same as used with the aluminum piston. We followed our regular practice of bringing the top ring well down from the top of the piston.

(c) We did not find it necessary or advisable to use a heat dam on the iron piston.

(d) We did not lower the compression ratio with the iron piston. It apparently cooled satisfactorily. With 70-octane gasoline there has been no indication of detonation or distress due to detonation.

(e) Pistons do not have a balancing ring at the bottom. Pistons are selected in grades, comprising six different weights, and matched within ¼ oz in sets.

(f) The iron piston is ground with the same cam with which we grind the aluminum piston, which gives a diameter of 0.010 in. less, parallel to the wristpin. The piston is ground straight, with no taper. The piston is tinned with 0.001 to 0.0012 in. of tin (diameter 0.002 to 0.0024 in.).

(g) Piston Rattle – When we first used this iron piston, we fitted it with a feeler 0.0015 in. thick and ⅜ in. wide. We found on the road a peculiar rattle which came in at speeds up to 35 mph after the engine was hot. It was very difficult to trace the cause, and we first ascribed it to wristpin fit but, after a long period of testing, we eliminated the idea of the difficulty being in the wristpin fit and found that it was the piston fit. We then fitted the pistons without any shim at all so that they would just fail to fall of their own weight in the piston bore. This arrangement immediately solved the piston rattle and we did not find any bad effects from this close fit. We found that the wristpin could be as loose as 0.0005 in. in the wristpin bronze bushing without any wristpin knock.

6. Oil Consumption

The oil consumption with the iron pistons has been amazingly good. At 30 mph we have been unable to measure any oil consumption and, at 50 mph, it is about 8000 mpg. The oil does not deteriorate with test mileage up to at least 25,000 miles, the distance which we have run so far. There is no indication whatsoever that the engine is running too dry.

7. Horsepower

Horsepower curves showing comparison between the engine output with aluminum and with cast-iron pistons indicate that, up to 4000 rpm, there is very little difference in performance. Above 4000 rpm, two possible factors enter into the picture:

(a) Increased internal engine friction due to rapidly increased inertia loads of the iron pistons.

(b) The heating up of the iron piston in the center, thereby reducing the volumetric efficiency of the engine in a constantly increased amount above 4000 rpm.

We have no way of separating these losses for individual evaluation.

8. Bearing Loads

The following table shows the maximum bearing loads with aluminum and cast-iron pistons at three speeds:

	3900 Rpm		4400 Rpm		4800 Rpm	
	Al.	C.I.	Al.	C.I.	Al.	C.I.
	psi		psi		psi	
Piston Pin	2874	2874	2874	2874	2874	2874
Crankpin	1600	2060	2040	2620	2430	3120
Front Main	790	1020	1010	1290	1200	1540
Center Main	1220	1570	1560	2000	1850	2380
Rear Main	770	770	980	980	1160	1160

The foregoing bearing loads are based on the following data:

	Al.	C.I.
Weight of piston, piston pins, and rings, lb.	1.094	1.937
Effective projected area, piston pin, sq in.	1.294	
Effective projected area, crankpin, sq in.	2.104	
Effective projected area, front main, sq in.	3.752	
Effective projected area, center main, sq in.	4.011	
Effective projected area, rear main, sq in.	3.866	

Weight of connecting-rod and big-end liner – total 2.20 lb – upper end, 0.55 lb; lower end, 1.65 lb.

9. Engine Smoothness

The use of the iron pistons quite noticeably "roughed-up" the engine in the lower ranges of speed but, strangely enough, at the upper ranges this roughness was not so noticeable. We ascribe this apparently peculiar behavior to engine friction, because after 5000 miles the tendency is to become quite smooth and we lose very little in smoothness in the long run.

10. Piston Surfaces

We find that tinned pistons are less subject to scoring and scuffing than are phosphate-treated pistons.

■ Piston Design "B"

General Notes –

At the start of 1942 production, cast-iron pistons were used in our $3\frac{1}{2}$-in. bore engine and cast-steel pistons in the $3\frac{1}{4}$-in. bore engine. Cast-steel pistons were released later for use in both engines after the necessary development work had been completed and the foundry had built up sufficient facilities to handle the entire production.

The principal objection to the use of cast iron as a piston material was the excessive weight of pistons having satisfactory strength. Experimental lighter designs of cast-iron pistons had resulted in the head pulling off at the lower ring groove while running at high speeds; also, the pin bosses cracked where the fillet joined the skirt. After these weak sections had been corrected with heavier walls and sufficient ribbing around the pin bosses, the resultant weight of the cast-iron piston for the $3\frac{1}{2}$-in. bore engine was 2.1 lb, 190% heavier than virgin-aluminum pistons. Due to the excessive weight of the cast iron, development work was concentrated on the use of a cast steel, which has approximately three times the strength of cast iron. Facilities for making cast-steel pistons had not been available at the start of the development program. The increased strength of the material allowed a substantial reduction in wall thickness throughout the piston, as well as lighter ribbing. The resultant weight of this piston was 1.7 lb, as compared with 2.1 lb with the cast iron.

The cast-steel piston, as originally designed and released for the $3\frac{1}{4}$-in. bore engine, gave very little trouble. It was found, however, that there was an appreciable increase in combustion roughness when the piston head thickness was less than 0.130 in.

The foundry technique for the production of steel piston castings has not reached its ultimate refinement as considerable variations occur in production castings. When this technique has been improved, we believe that a further reduction in weight will be possible by reducing wall thickness of the skirt.

The comparative weights of the aluminum, cast-iron, and cast-steel pistons are:

Piston Design "B" – Weights of Aluminum, Cast-Iron, and Cast-Steel Pistons

Piston Material	Weight, lb	Ratio $\frac{\text{Weight}}{\text{Virgin Al. Weight}}$
$3\frac{1}{2}$-In. Bore Engine		
Virgin Aluminum	1.09	1.00
Secondary Aluminum	1.14	1.04
Cast Iron	2.1	1.9
Cast Steel	1.7	1.6
$3\frac{1}{4}$-In. Bore Engine		
Virgin Aluminum	1.00	1.00
Secondary Aluminum	1.05	1.05
Cast Steel	1.5	1.5

Design Features

In general design, both sizes of pistons are circular-ground with a full skirt having a relief ground around the pin. The ring combination consists of two 3/32-in. compression rings and two 3/16-in. oil rings, all above the pin. The pistons are tin-plated at the present time. Wall thickness of the cast-steel skirts is 0.042 to 0.050 in., and the head thickness, 0.146 $\pm$ .005 in. on the $3\frac{1}{2}$-in. piston and 0.156 in. $\pm$ .005 in. on the $3\frac{1}{4}$-in. piston.

Performance

From the standpoint of performance, both $3\frac{1}{2}$ and $3\frac{1}{4}$-in. bore engines have somewhat higher friction with cast-steel than with aluminum pistons. The increased friction is due to the circular-ground pistons which we found necessary to fit from 0.00125 to 0.00175 in. clearance, or to have a 7 to 20 lb pull on a 0.0015 x $\frac{1}{2}$ in. ribbon. The clearance with aluminum pistons was 0.0015 in. at the bottom of the skirt. However, they were cam-ground and had a reverse taper, that is, smaller at the top of the skirt than at the bottom. Cam-grinding iron pistons has shown a reduction in friction; however, early experimental work indicated that the cam-ground pistons were more sensitive to piston slap than are circular-ground pistons.

Iron pistons induce an increased amount of combustion roughness.

It is necessary to hold closer bearing clearances to eliminate bearing rattle.

No material difference in car performance and economy has been observed due to iron pistons.

Oil economy is as good or, on an average, somewhat better with iron than with aluminum pistons.

The $3\frac{1}{2}$-in. bore engine is somewhat rougher at extremely high speeds with iron pistons.

Changes in bearing materials have made it unnecessary to sacrifice the general endurance life of the engine.

Other Changes Made When Using Iron as a Piston Material

1. Piston-pin bushings had to be changed to a non-corrosive material.

2. Piston pins were strengthened by increasing the wall thickness 1/64 in. and lengthening the pin $\frac{1}{8}$ in. to give more bearing area and better support in the piston.

3. The babbitt main and connecting-rod bearings were replaced with a material having a copper-nickel matrix supporting a 0.002 in. overlay of babbitt. This bearing has much more strength than one of babbitt.

4. The connecting-rod forging was strengthened by increasing the width of the I-beam section 1/16 in.

5. The crankshaft of the $3\frac{1}{4}$-in. bore engine was stiffened by using narrower rod bearings. The change in the crankshaft was made to improve main bearing life.

The foregoing statements are a brief summary of our experiences encountered in the development of cast-iron and cast-steel pistons. The following tables give further comparative information:

Summary of Cold-Room Cranking Tests at 0 F

	Cranking Rpm 10-W Oil – 8700 Saybolt sec
$3\frac{1}{2}$-In. Bore Engine	
Cam-ground aluminum pistons fitted 0.00175-in. clearance	43.9
Circular-ground steel pistons fitted 0.0015-in. clearance	42.2
$3\frac{1}{4}$-In. Bore Engine	
Cam-ground aluminum pistons fitted 0.002-in. clearance	47.0
Circular-ground steel pistons fitted 0.0015-in. clearance	42.2

Analysis of Cast-Steel Piston Iron, %

Carbon	2.6-2.85 (all combined)
Silicon	1.2-1.45
Manganese	0.37-0.43
Sulfur	Under 0.15
Phosphorus	Under 0.15

■ Piston Design "C" (See Fig. 2)

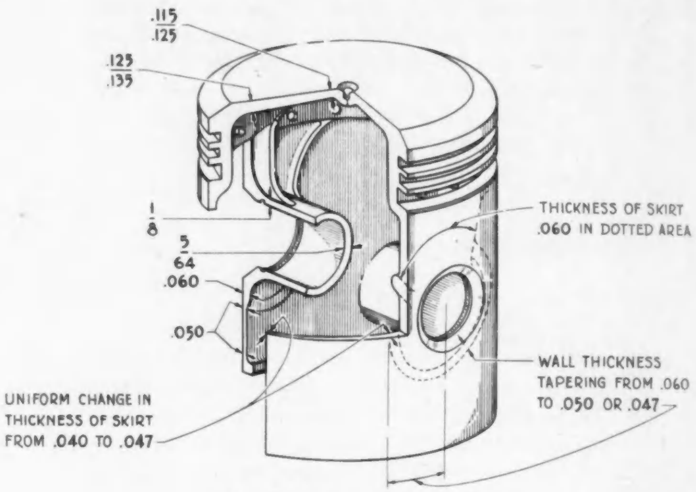
The following is a brief statement of our experience in the development of iron pistons for our engines:

Having examined several types of cast-iron pistons which had been in use for some time and finding a big variation in the designs, the reasons for which were obscure to us, we decided to start on our own, using the best logic we could muster. It was our thinking that, in view of bearing loads and other considerations, an effort should be made to approach the aluminum weight as closely as possible in the first attempt and try to uncover defects or points of indicated failure, applying corrections as we proceeded.

Since we found many different types of relief around the pin hole areas, we decided to start with a standard oval grind which, if acceptable, could be obtained with existing equipment. This grinding has not had to be changed other than to make changes in the amount of ovality.

We ran the first pistons with both tin plating on the surface and with the phosphate treatment. We found the phosphate finish preferable to the tin, and much easier to process.

The original design had a continuous rib from the top of the pin bosses up to and across the piston head. Early in the experience a sidewise movement of the piston-pin bosses was indicated by high points of bearing above the pin boss on the high thrust side; also, a rapid wearing of the piston pin in the bosses where originally the pin was allowed to float. Next, the diagonal ribs leading from the pin boss to the ring belt were added and the center rib removed. This was done without adding weight. It was then found that the lateral weaving of the boss was stopped but that a vertical breathing had begun which resulted in cracks beginning below the lower ring in line with the boss, some of which extended up into the ring belt. Next, the center rib was added again, but all ribs were made thinner and not quite so high, resulting in the addition of about 0.3 oz of weight. At the same time skirt thickness in the area adjoining the pin boss was increased slightly and a former balancing boss at the end of the skirt reduced,



■ Fig. 2 – Piston Design "C"

bringing the weight back to the original. These last changes removed all indication of breathing or deflection, even when tested running on the torque peak with excess spark advance.

In all of this work we were constantly experiencing excess pin wear in the pin bosses. It was decided that the pin must be prevented from turning in the piston. This was accomplished by making the pin a very tight fit in one boss and a normally snug fit in the other boss. This was found to be quite satisfactory.

Another objection consistently present was that of smoking from the crankcase. We determined that this was not blowby but actual smoke from the burning of oil under the head. We tried increasing the head thickness, which made the condition worse. Then it was reasoned that, if more oil were washed over the head, the metal temperature might be reduced below the smoking point. Already having rifle-drilled rods, we put a concentric groove in the piston-pin bushing and drilled a squirt hole out of the top of the rod, then lengthened the registration groove in the lower bearing; and, after some experimentation using a piston with a head cut off to observe the oil delivery at various speeds, we tried this system under operating conditions. It was found that the smoking completely disappeared, as did the accumulation of carbon under the head, and that the compression ratio to produce incipient knock could be increased 0.3.

We were successful in maintaining the originally anticipated minimum weight for both sizes, which made it un-

necessary to go into bearing design changes. The connecting-rod bolt diameter was increased 1/32 in., however, in the smaller engine.

The piston rings are identical with those used in the aluminum pistons, except that the side clearance on the compression rings was increased slightly.

The outer edge of the top land was beveled originally to reduce weight without moving the rings any nearer the top end of the bore. In fact, the volume represented by this change in shape also represents the only difference in compression ratio between the engines using iron and aluminum pistons.

The material used is a conventional cast iron – not an electric-furnace iron. Some defective castings have been had and a few had shown up after being in engines before more rigid inspection methods were introduced.

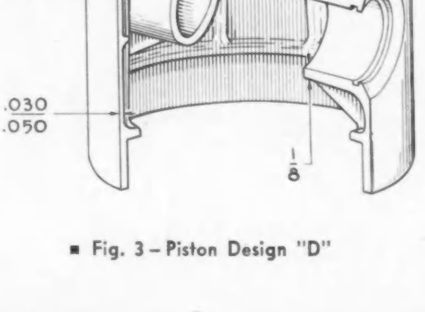
To date the life history of this design has been very satisfactory and, in all cases, the oil mileage averages better than with aluminum pistons. The weights are as follows:

3⅜-In. Bore	Aluminum	1.12 lb
	Cast Iron	1.23 lb
3⅛-In. Bore	Aluminum	0.95 lb
	Cast Iron	1.14 lb

In conclusion, it is our belief that, in many instances, failures of cast-iron pistons have resulted as much from too much "beefing up" as they have from not enough. As in other design, if concentration of stresses can be located and removed, the job is well done without adding extra metal.

■ Piston Design "D" (See Fig. 3)

When it was decided to develop iron pistons for our passenger-car engines, the various cast-iron piston designs were studied as to their relative merits. The design finally selected for development was the lightest design, which had previously been tested on our engines and had given satisfactory results.



■ Fig. 3 – Piston Design "D"

This piston design has six vertical ribs and one horizontal rib in the skirt plus a vertical rib from the pin boss to the head. The skirt is quite thin, being about 0.030 to 0.040 in. with a large share of the strength coming from the ribs. A design was brought through using our regular aluminum piston-ring band line-up, shortening the piston, and adding a continuous balancing ring at the bottom of the skirt. The original balancing ring was short and thick but later, in production it was made longer and thinner to provide greater accuracy in the balancing operation and to provide more balancing material.

In the initial tie-up and cold-scuffing tests a comparison was made between tin plate and one of the phosphate coatings. It was found that the phosphate coating compared very favorably and, consequently, all further tests were made with this coating because it was felt that, if the necessity arose to go to cast-iron pistons, tin also would be very scarce.

In surveying other piston designs, it was found that the cam-ground pistons were generally used. Consequently, initial tests were made with cam-ground pistons. However, the aluminum piston cam was used which did not provide very good bearing characteristics and which gave very poor oil economy. Consequently, the cam contour was changed to a round thrust face for approximately 50 deg on each side, dropping off at the pin bosses to 0.003 to 0.004 in.

This change improved the oil economy considerably and brought it within commercial limits.

As a result of numerous tie-up, power, cold-slap, cold-scuffing, and endurance tests, and oil economy checks, the following grinding practice was established:

Cam: 0.003-0.004 in.

Skirt taper: (large diameter at bottom of skirt) 0.001-0.002 in.

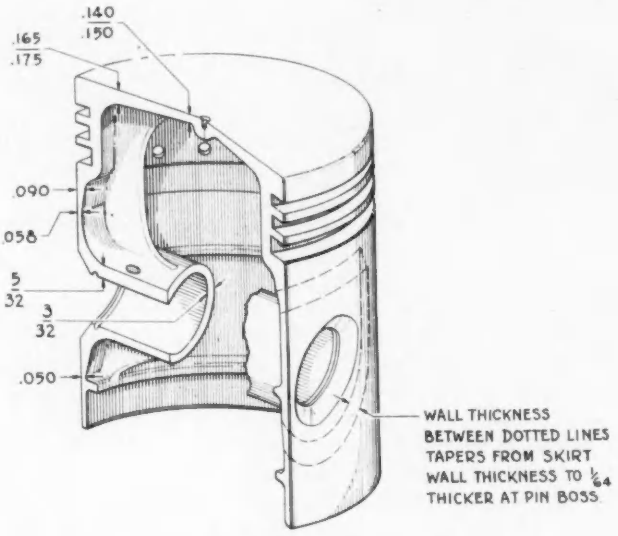
Sufficient clearance to give 10 to 15 lb pull on a ½-in. wide by 0.0015 in. thick feeler stock. This amount of feeler pull results in 0.0008 to 0.0018-in. clearance at the center of the piston skirt. Some of the first mechanical trouble with the iron pistons was experienced with piston pins, because when full-floating pins were used, running directly on the phosphated surface, occasional galling of the wristpin would result. It was found that, by improving the finish of the pin hole and boring it 0.0004 in. larger than the desired finished size to allow for build-up of the phosphate coating, the trouble was overcome. If this allowance for clearance was not made, the pins did not turn in the piston, thus overheating the pin sufficiently to gall the bushing in the rod.

Along with this change, the pins were fitted slightly looser in the connecting rod as well. It was found that bushings in the piston-pin bosses were not satisfactory because they would rotate in the piston and gall the pin.

On high-speed endurance runs it was found that the piston heads sometimes pulled off through the lower ring groove, and that the skirts would crack around the pin bosses. To overcome these troubles, the cross-section of the ring belt, the skirt thickness around the pin boss, and the radius of the fillet joining the pin boss to the skirt were increased. Also, the ribs on the four sides of each pin boss adjoining the skirt were made larger.

After making the numerous changes that were necessary to produce satisfactory pistons from an endurance standpoint, more complete oil-consumption runs were made. It was found that the oil consumption of some engines was very good, whereas with others it was very poor. It ap-

peared that most of the trouble was caused by seating of the rings, as usually the rings in the poorer economy engines would be glazed and show no wear-in characteristics. Improvements in the oil economy were obtained by modifying the piston-ring shape, cam contour, and oil-ring tension. After making all the changes in experimental and production pistons, the following weights were established, as compared with aluminum:



■ Fig. 4 – Piston Design "F"

	$3\frac{1}{4}$-*In. Bore*	*3 7/16-In. Bore*
Aluminum Piston	0.99 lb	1.16 lb
Cast-Iron Piston	1.67 lb	1.89 lb

Metallurgical Problems –

In order to obtain light cast-iron pistons, it is necessary to maintain rather accurate control of the inside diameter of the casting. Our pistons were cast in green sand molds with dry sand cores, and considerable difficulty was encountered in the foundries in obtaining a uniform size to these cores. It was found desirable to gage the cores 100% in order to obtain sufficiently good control of the piston wall thickness. The first experience in machining cast-iron pistons indicated a wide variation in the machinability of the pistons as received. Considerable difficulty led to the requirement of a maximum Brinell hardness of 190. Though the harder pistons caused difficulty in machining, pistons were obtained which were so soft as to be dangerous to use. In our engine testing it was found that pistons under 149 Brinell were unsafe to use. Pistons received from the foundries in the beginning were found to vary from as low as 112 Brinell to as high as 248. The cause for the extremely soft pistons was traced mainly to the practice of shaking the pistons out of the molds while very hot and placing them in large tote boxes. The center pistons in these boxes cooled very slowly and became completely annealed, whereas those at the top and edges were scarcely affected. As a result of our experience, a specification was written requiring that pistons, as cast, be not less than 180 nor over 223 Brinell.

The composition of these pistons was within the following range:

Total Carbon	3.20 to 3.50%
Manganese	0.50 to 0.75%
Silicon	2.10 to 2.40%
Sulfur	0.125% Maximum
Phosphorus	0.10 to 0.25%

This iron cast into standard 1.2 in. arbitration bars after annealing will have a minimum strength of 25,000 psi.

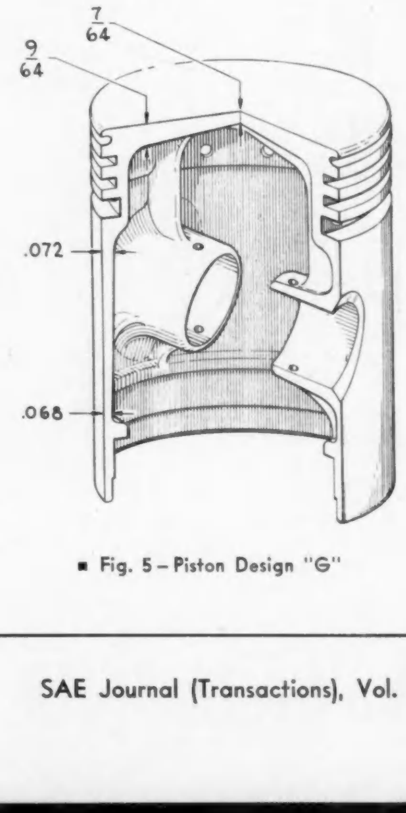
It was found desirable to brinell the pistons 100% after annealing, and this brinelling operation greatly reduced the machining difficulty encountered in our early experience.

■ Piston Design "E"

The following summarizes briefly our experiences in developing cast-iron pistons. These pistons were designed along the same lines as the cast-iron pistons used a few years back, before changing to aluminum, except that we decided to retain the piston-ring line-up used in the aluminum pistons.

In view of the short time available for the change-over, it was necessary to use the same manufacturing line-up used for aluminum pistons and, consequently, it was necessary to develop cams for the grinding machines which would produce a piston shape giving satisfactory contact. The final contour is symmetrical and drops 0.00002 in. at 5 deg; 0.00176 in. at 45 deg; .003 in. at 75 deg; and 0.003 in. at 90 deg.

As can be imagined, considerable difficulty was encountered in the foundry since the shape of the top of the piston cannot be readily machined and, consequently, must be held in a rough casting to very close limits for shape and



■ Fig. 5 – Piston Design "G"

thickness in order to avoid excessively large weights for balancing. The production pistons are held to a specified fit of $\pm 1/16$ oz and a maximum of 2 oz of cast iron is allowed in the balancing ring.

The volume variation between piston heads is held to ± 0.06 cu in. Piston fits are held to 0.0014 to 0.002 in. Piston weights are: cast iron, 1.55 lb, and aluminum 0.91 lb. The oil consumption with the iron pistons has remained practically the same as with aluminum pistons, using the same rings.

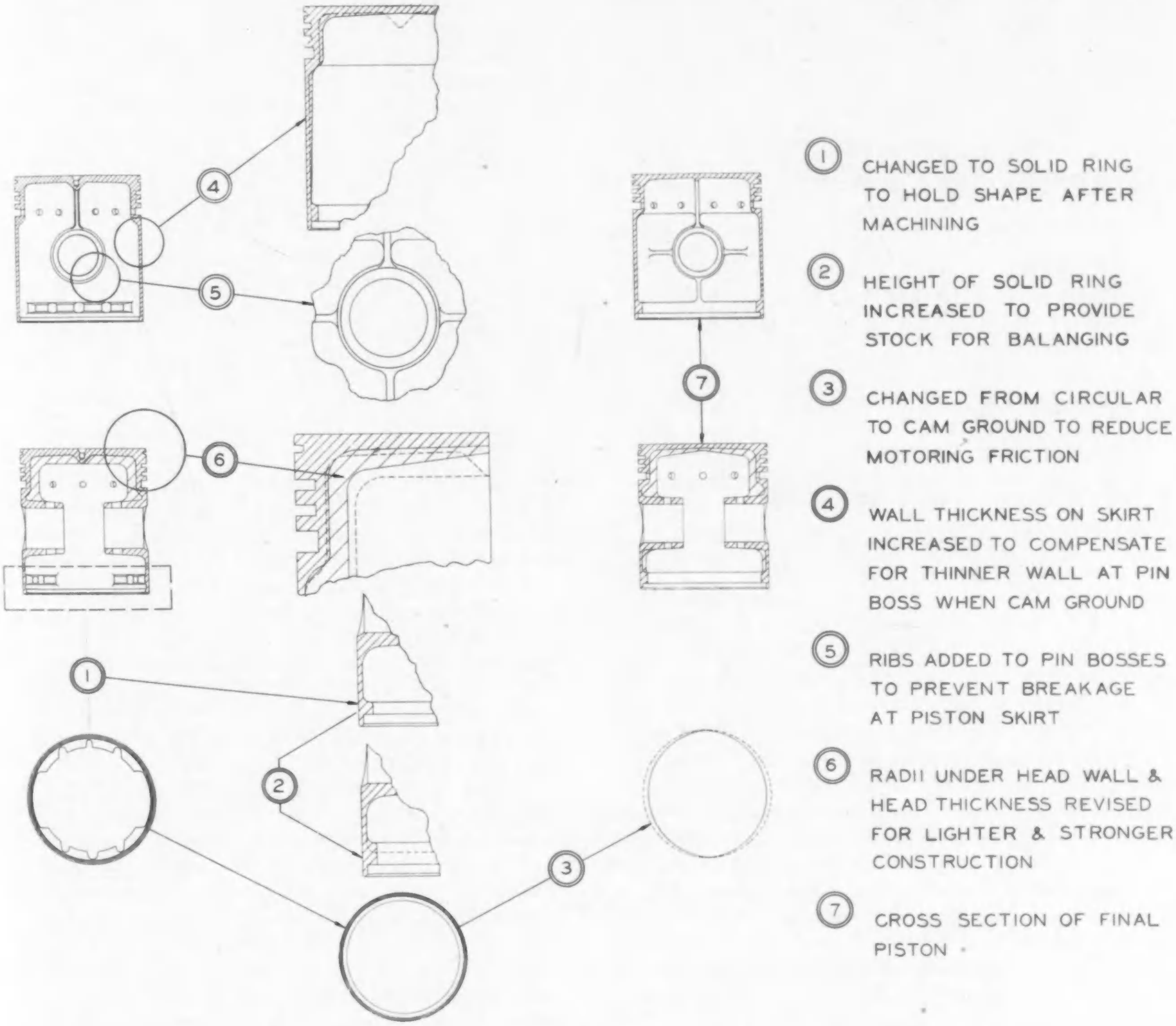
Due to the higher operating temperatures of the cast-iron piston heads, it was necessary to decrease compression ratio 0.4. Piston-pin inertia loads were increased 50% and the crankpin and main bearing loads were increased 26%. Due to the development of a special finish for the crankpins and main bearings, the bearing life, despite the increased loading, has remained practically the same as with the aluminum pistons. The increased strength in the connecting rods was obtained by changing the material from SAE 1045 to X-1345 and shot-blasting the rods.

No cooling problems were encountered since the total heat rejection did not change appreciably. While the rejection to the head jacket water increased, due to the lowered compression, the rejection in the block decreased due to the lowered thermal conductivity of the iron pistons.

■ Piston Design "F" (See Fig. 4)

The principal changes in design during cast-iron piston development were as follows:

The first design of cast-iron piston was cam-ground and incorporated a continuous balancing ring at the bottom of the piston. Operation was satisfactory but, because of excessive weight, the wall section was reduced slightly and



■ Fig. 6 – Principal design changes during the development of cast-iron piston "G," shown in Fig. 5

the tapered section below the ring band removed. This change resulted in failure at the wristpin boss at high speed so the wall thickness was re-established to its former value, a rib added from the pin boss downward to the balancing ring, the balancing ring height reduced, and a 1/64-in. circular reinforcement about the pin boss added.

This design resulted in failure of the piston head pulling off at the upper ring groove. A taper reinforcement was then added below the ring groove; the rib tying the pin boss to the roof was increased ⅛-in. in width; and the wall thickness behind the ring groove was also increased. These changes gave a satisfactory piston which was approved for production. No anti-scuff coating was found necessary.

The weight of the cast-iron piston is 1.55 lb.

The corresponding weight of aluminum piston is 1.06 lb.

The increase in load on the connecting rods at 4500 rpm, with the use of cast-iron pistons, was 10.3%.

The average oil mileage on a 1000-mile run with SAE 20 oil was 3600 mpg with aluminum pistons, and 4300 mpg with cast-iron pistons.

The cranking speeds at zero F, and using SAE 10-W oil, were as follows:

Pistons	Cranking Speeds, rpm
Aluminum, cam-ground	33
Cast-iron, cam-ground, phosphate-coated	41
Cast-iron, cam-ground, uncoated	44

The chemical analysis of the cast iron used is as follows:

Total carbon	3.20-3.55
Silicon	2.20-2.50
Manganese	0.50-0.90
Phosphorus	0.15-0.35
Nickel	0.25 max.
Chromium	0.25 max.
Brinell	156-183

■ Piston Design "G" (See Fig. 5)

The principal changes in design during the development of our cast-iron pistons are illustrated in Fig. 6. The original sample cast-iron pistons were circle-ground and incorporated a balancing ring scalloped in a manner similar to that previously used on aluminum pistons. The Manufacturing Department requested that the balancing ring be made continuous, in order to provide additional stiffness for machining. This balancing ring was later changed as shown at (2), to give a greater amount of balancing material, and the inner diameter was machined to insure a uniform thickness. The stiffer and uniform-section balancing ring showed a marked improvement in the machinability of the pistons, it being possible in production to hold both the taper and the contour of the piston very accurately.

The original circle-ground pistons were tin-plated, and sufficient experimental work was completed to indicate that they would prove satisfactory for production use. However, tests in the cold room indicated that the cranking speed was too low for satisfactory starting. Cranking-speed tests showed that satisfactory starts could be obtained if the pistons were cam-ground instead of circular-ground. Because of inability to obtain tools, the same cams were used for the cast-iron pistons that had been used for aluminum pistons. A summary of the results of cold room tests is given in the following table:

Summary of Cold-Room Cranking Tests made at 0 F

	Cranking RPM	
	SAE 10 Oil – 6700 Sec Saybolt	SAE 20 Oil – 45,000 Sec Saybolt
3-In. Bore Engine		
Aluminum pistons – fitted 16-21 lb pull on 0.002x1-in. shim	45.5	23
Circular-ground C. I. pistons fitted 10-15 lb 0.002x1-in. shim	37	24
Cam-ground C. I. pistons fitted 10-15 lb 0.002x1-in. shim	44.5	27
3 5/16-In. Bore Engine		
Aluminum pistons – fitted 14-19 lb pull on 0.002x1-in. shim	39	24
Circular-ground C. I. pistons fitted 10-15 lb 0.002x1-in. shim	26	13.5
Circular-ground C. I. pistons fitted 10-15 lb 0.003x1-in. shim	32.5	17
Cam-ground C. I. pistons fitted 10-15 lb pull on 0.002x1-in. shim	40	22
Cam-ground C. I. pistons fitted 17-20 lb pull on 0.002x1-in. shim	37	18.5
3 1/16-In. Bore Engine		
Aluminum pistons – fitted 7-12 lb pull on 0.003x1-in. shim	55	40
Circular-ground pistons fitted 10-15 lb pull on 0.002x1-in. shim	36.5	20
Cam-ground pistons fitted 18-22 lb pull on 0.002x1-in. shim	31.3	...
Cam-ground pistons fitted 10-15 lb pull on 0.002x1-in. shim	42.0	23

About the time the necessity for cam grinding was discovered, a phosphate type of anti-scuff coating was suggested as a possible substitute for tin plating. Cam-ground pistons were therefore tested with this method of treatment and, as it was found to be satisfactory, it was finally used in production.

The first mechanical failure encountered in the endurance testing occurred with circular-ground pistons. On rather short running the piston heads pulled off at the upper ring groove. This difficulty was later traced to an experimental three-piece pattern having a parting line at the point of failure. The failures, however, led to an examination of the section of the piston between the head and ring lands, and it was felt that this section was insufficient for either mechanical strength or heat transfer and, therefore, it was increased as shown at (6) on Fig 6. The second mechanical difficulty encountered was cracking around the wristpin bosses. These cracks developed in relatively short mileage with cam-ground pistons with thin walls, 0.050 in. or less, and changes (4) and (5) were made to correct this difficulty. Due to the practical completion of the patterns, it was not possible to make the inner diameter of the piston casting elliptical, so the thickness of the skirt wall was increased 0.008 in., as shown at (4), and three additional ribs were added, as shown at (5). The increased wall thickness and additional ribbing cor-

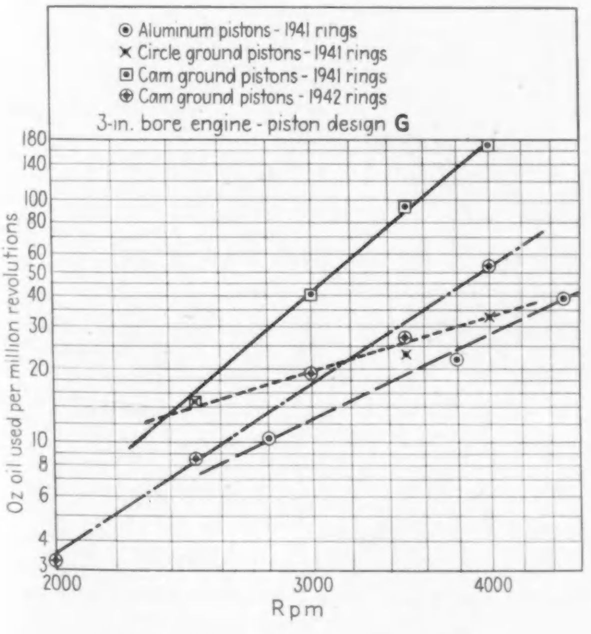
rected the failure around the wristpin bosses. The additional wall thickness and ribbing naturally increased the piston weight and were the cause for the increase in available balancing ring material previously mentioned. The final production weights of aluminum and cast-iron pistons are given in the following table:

Weights of Aluminum and Cast-Iron Pistons

Piston Material	Weight, lb	Ratio Weight / Virgin Al. Weight
3-In. Bore Engine		
Virgin Aluminum	0.560	1.00
Remelt Aluminum	0.590	1.05
Unribbed C.I.	1.18	2.11
Ribbed C.I.	1.26	2.25
3 5/16-In. Bore Engine		
Virgin Aluminum	0.91	1.00
Remelt Aluminum	0.95	1.04
Unribbed C.I.	1.63	1.80
Ribbed C.I.	1.75	1.92
3 1/16-In. Bore Engine		
Virgin Aluminum	0.86	1.00
Remelt Aluminum	0.89	1.03
Unribbed C.I.	1.41	1.64
Ribbed C.I.	1.53	1.78

Weights include piston-pin bushings. Piston-pin dimensions and weights remained unchanged.

Preliminary tests indicated a possibly serious increase in oil consumption with the use of cam-ground cast-iron pistons. In the case of the 3 1/16-in. bore engine, it was found necessary to fit the pistons tighter and, in all engines, a negative taper was used. By negative taper is meant that the diameter of the piston at the bottom of the skirt is greater than the diameter just below the ring belt. Satisfactory oil consumption was obtained by increasing the unit pressure on the oil rings from 20% in one engine to 150% in another. The unit pressure between the oil rings and the cylinder walls remained the same on all compression rings. This change in ring combination resulted in commercially satisfactory oil consumption. See Fig. 7.



▪ **Fig. 7 – Oil used per million revolutions**

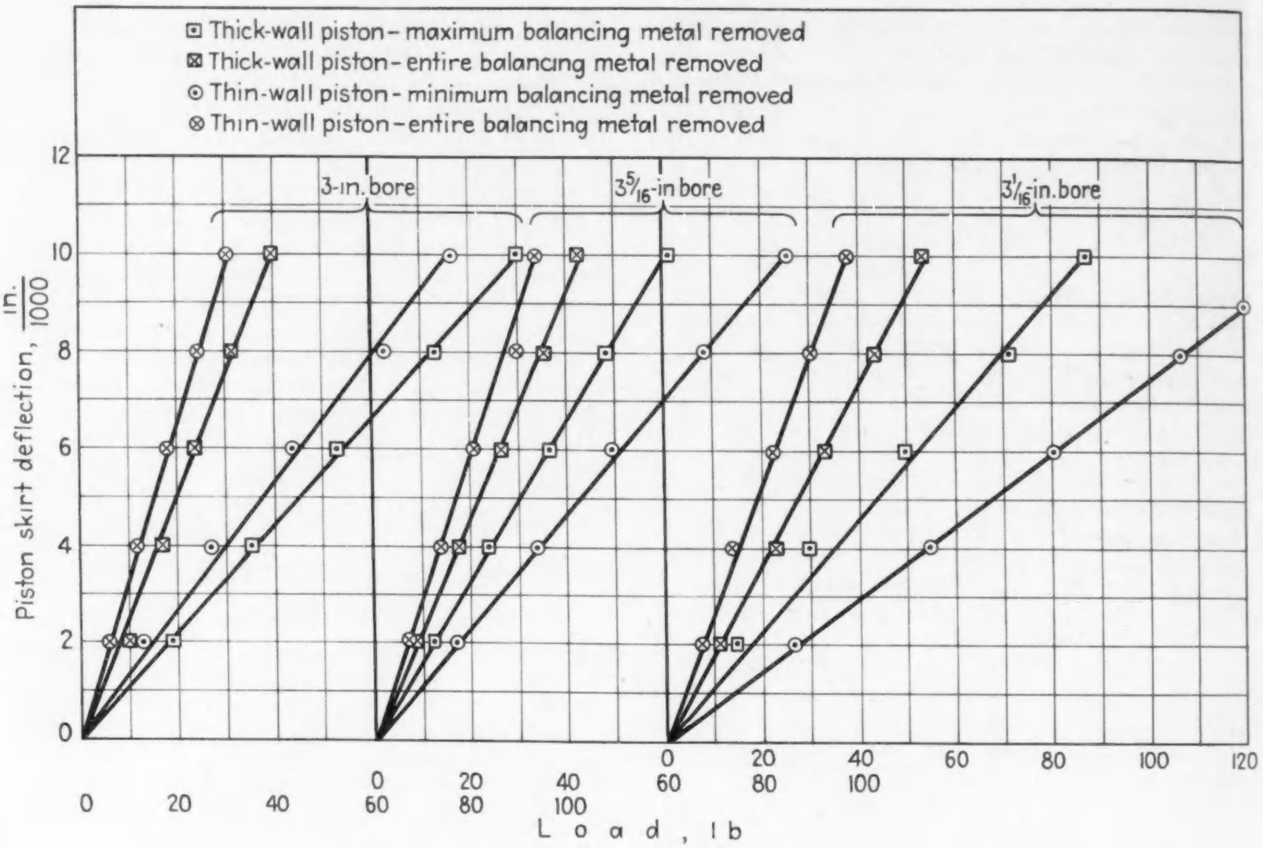
Although the final weights of the cast-iron pistons were from 70% to 100% heavier than the aluminum pistons, the percentage increase in weight on the small end of the rod was much less. It was decided that time did not permit the development of an extremely light cast-iron piston or a bearing material of materially greater life than those at present known and that the best way to compensate for the increased piston weight was to reduce engine speeds. Axle ratios were, therefore, arbitrarily reduced approximately 10%. This arrangement left the bearing loads in two of our engines the same and increased them slightly on the other engine, as shown in the following table:

Percentage Increase in Bearing Loads with Use of Cast Iron in Place of Aluminum Pistons

	Percentage Increase in Bearing Loads at 4500 Rpm with Cast-Iron Pistons		Percentage Change in Bearing Loads with Aluminum Pistons at 4500 Rpm and Cast-Iron Pistons at 4000 Rpm	
Engine	Rod Bearings	Main Bearings	Rod Bearings	Main Bearings
3-In. Bore	34½%	27½%	6½%	1½%
3 5/16-In. Bore	25½%	19.0%	0	0
3 1/16-In. Bore	21.0	15.0	0	0

The difficulties encountered in production were relatively few. With the lighter unribbed pistons some difficulty was encountered in holding the proper taper and cam contour. This difficulty disappeared as soon as the wall was increased in thickness and the ribs added at the wristpin boss. At the outset of production, three weights of pistons were required because of insufficient weight in the balance ring. The amount of balance weight was increased so that at present two sets of weights are used on all pistons. Some difficulty was noticed in the shop because of slower machining time on the cast-iron pistons. It was necessary to take precautions to prevent burring or nicking at the lower edge of the piston skirt, as nicks at this point made accurate determination of feeler pull impossible. In short mileage the nicks would wear down and the pistons would be too loose. The usual minor difficulties with blowholes and cooling strain cracks were encountered.

The method used in fitting the pistons is by specifying the pounds pull on a strip of shim stock, 1 in. wide and 0.002 in. thick, placed between the piston and the bore. With a negative taper on the piston, the characteristics of the piston, when considered as a spring, affect dimensional fits between piston and bore. It is obvious that, with an unsplit piston skirt, the stiffness of the piston in compression across the thrust faces will be affected by the amount of material left in the balancing ring and also by the thickness of the piston skirt wall. Some data on this characteristic of pistons are given in Figs. 8 and 9.



■ Fig. 8 – Deflection at bottom of piston skirt versus load

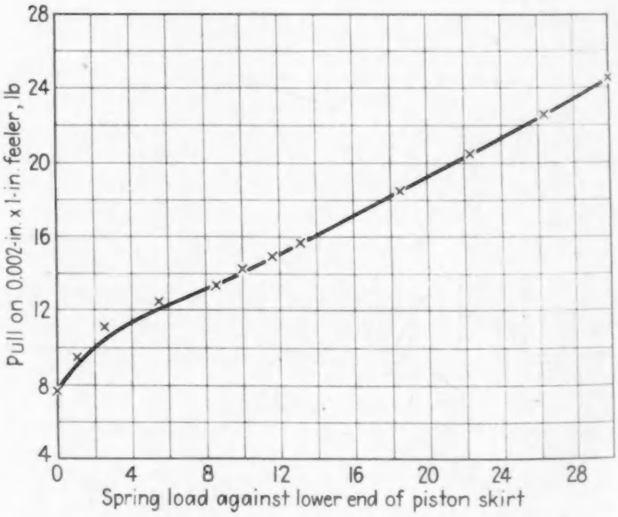
The chemical analyses of two samples of piston cast iron are given as follows:

	Sample No. 1	Sample No. 2
Total carbon	3.22	3.36
Silicon	2.83	2.27
Sulfur	0.075	0.084
Phosphorus	0.185	0.175
Manganese	0.68	0.70
Chromium	Trace	Trace
Nickel	Trace	Trace

■ Editorial Comment

From these experiences it appears that such problems as oil consumption, detonation, engine friction, piston scuffing, piston slap and rattle, and so on, were not particularly serious when changing from aluminum to cast-iron pistons. The most serious problem was the increase in loads on the bearings and stresses in the connecting-rod and piston pins due to the greater weight of the cast-iron pistons.

Table 1 gives the ratio of the weights of cast-iron pistons to equivalent aluminum pistons and also the maximum amount of weight found necessary to provide for possible



■ Fig. 9 – Steady pull on 0.002 x 1-in. feeler versus spring load on lower edge of piston skirt

removal in order to balance a run of production pistons to the same weight. It will be noted that, on the average, the cast-iron piston is 65% heavier than the equivalent aluminum piston, but the designs described varied from 9% to 125% more than the aluminum weight. The weight of a given volume of cast iron is 165% greater than the weight of a similar volume of aluminum. The figures giving the percent of the piston weight necessary to permit balancing to a single weight indicate a possible variation of from 5% to 10% in the uniformity of cast sections with present foundry practice. As a matter of interest, the weights of 17 aluminum and 13 cast-iron and steel pistons were computed in ounces per cubic inch of the cylinder bore volume included in the piston length. The aluminum pistons varied from

$$0.43 \text{ to } 0.61 \text{ oz/in.}^3$$

and the cast-iron pistons varied from

$$0.56 \text{ to } 1.14 \text{ oz/in.}^3$$

The averages were about

0.50 oz/in.^{3} for aluminum, and
0.85 oz/in.^{3} for cast iron.

Conclusion

The principal problem in the use of cast iron for pistons at this time is to design for low weight and adequate strength.

In the last analysis, the design of a cast-iron piston is controlled by the ability of the foundry to cast thin sections to close tolerances. As this part of foundry technique improves, cast-iron pistons can be made lighter, and possibly as light as the present designs of aluminum pistons.

DISCUSSION

Construction of Additional Piston Type

—Arthur Townhill
Thompson Products, Inc.

THE pistons mentioned in Mr. James' paper indeed show a lot of good engineering and a surprising amount of success in equaling the weight of the aluminum piston. There is one type which, theoretically at least, seems quite promising and I believe should have further development. This is a type which has a construction very similar to that of the Ray Day piston which is made in aluminum, except that the skirt is not separated from the head.

Continental and Oliver Hart Parr both use this type of piston in some of their models. In order to make it clear to some of those who have not seen these pistons, the pin boss is supported by ribs running from the head to the inner edge of the pin boss transverse of the wristpin axis, and also by the ring belt and skirt over the boss. The reason for believing in this construction is twofold:

1. The load is carried directly from a point near the center of the head to the inner edge of the piston-pin boss, hence to a point within the skirt of the piston. Inasmuch as many of the pistons of the type which have been described have failed in the skirt section around the pin boss during their development, it indicates that a great portion of the load has been caused by the bending of the wristpin. This suggested design would distribute this load on four points across the wristpin. Two points would be at the diameter of the skirt section, and two points would be at the inner edge of the piston-pin boss.

2. The heat which normally would flow from the center of the head toward the ring groove would be somewhat bypassed down these two ribs, shortening the path of travel and lowering the tendency to detonate. This type is not made from the simplest type of mold, but I have noticed that considerable more money has been spent on the aforementioned type than hitherto, to produce lightness. Some increase in cost can now be absorbed by gaining in lightness. I believe refinement in this design would produce the lightest and strongest type.

Table 1 - Summary of Comparative Data

Piston Design	Ratio Wt. Iron or Steel Piston / Wt. Al. Piston	Piston Wt. per Cu. In. of Cylinder-Bore Length. Al.	C. I.	Max. Weight Removable for Balancing, % Total Weight	Length of C. I. or Steel Piston / Length of Al. Piston
"A" $3\frac{1}{8}$-In. Bore	1.77	0.61	1.14	....	0.95
"B" $3\frac{1}{2}$-In. Bore (Cast Iron)	1.93	0.45	0.93	....	0.93
$3\frac{1}{2}$-In. Bore (Cast Steel)	1.56	0.45	0.75	....	0.93
$3\frac{1}{4}$-In. Bore (Cast Steel)	1.50	0.49	0.71	....	0.98
"C" $3\frac{3}{8}$-In. Bore	1.09	0.52	0.57	....	1.00
$3\frac{1}{8}$-In. Bore	1.20	0.59	0.70	....	1.00
"D" $3\frac{1}{4}$-In. Bore	1.68	0.52	0.92	8	0.95
$3\frac{7}{16}$-In. Bore	1.64	0.49	0.97	7	0.90
"E" $3\frac{3}{32}$-In. Bore	1.71	0.45	0.77	8	....
"F" $3\frac{1}{4}$-In. Bore	1.42	0.54	0.81	11	0.95
"G" 3-In. Bore	2.25	0.43	0.99	5	1.00
$3\frac{5}{16}$-In. Bore	1.92	0.45	0.87	4	1.00
$3\frac{1}{16}$-In. Bore	1.78	0.46	0.81	7	1.00
Average	1.65	0.51	0.84	7%	0.97
Maximum	2.25	0.61	1.14	4%	0.90
Minimum	1.09	0.43	0.56	11%	1.00
Ratio Sp. Gr. Iron / Sp. Gr. Aluminum	2.64	....	....	....	....

LIGHT PLANE ENGINES and

THE recent revision of gasoline specifications is presenting a serious problem for designers of light airplane engines, where the customers are demanding increased power at no increase in cost or weight. The author points out the effects of using fuels of higher octane number containing greater proportions of lead, and describes attempts to alter the engine design to permit the use of higher leaded fuels without seriously increasing the weight or cost of the engines.

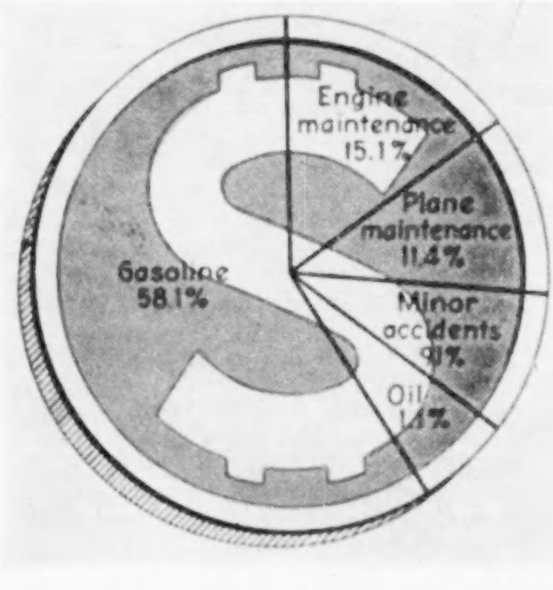
The use of dual carburetion, the author believes, on a 6-cyl opposed type of engine is the simplest method of approach for the time being. However, he points out that the designer of the light plane must better appreciate the necessity of better cooling and attention to details in the installation, in order to do his part in obtaining the maximum efficiency from the engine.

THE AUTHOR: CARL T. DOMAN (M '26), vice president and chief engineer, Aircooled Motors Corp., and secretary-treasurer of the Syracuse Section, took his first job with the Franklin Automobile Co. Many of the aircooled engine patents formerly held by the Franklin Automobile Co. and now owned by the Aircooled Motors Corp., are the inventions of Mr. Doman in collaboration with Edward S. Marks.

THERE seems to be no common definition for a light plane engine in terms of horsepower, as the present power range for the so-called light planes is found to be from 65 hp minimum to 175 hp maximum. For that reason this paper will attempt to discuss the light plane engines in a broad sense with particular emphasis placed on those engines from 65 to 175 hp, as the trend seems to be definitely toward higher and higher power requirements. Several months ago all the important light plane makers were approached for their definition of a light plane. A summary of their replies is given in the Appendix of this paper. Briefly, there are no two definitions which agree.

With the gradual increase in power requirements for light plane engines, low operating cost becomes more and more important. The importance of the operating costs is shown on Fig. 1 which gives an analysis of the hourly operating costs. This was taken from the work of Prof. Terry and Prof. Kellogg of Cornell University in their paper: "The Cost of Owning and Operating Small Aircraft."[1]

At the present time there are no gasoline airplane engines approved in this country for use with a fuel other than aviation grade. Many attempts have been made to use automotive gasoline in an airplane. However, these attempts have not been carried to a logical conclusion and the troubles eliminated from the installation, which condition prevents the use of automotive fuels. There must be a definite trend toward the automotive fuels as the light airplane becomes more and more popular for private use. As William B. Stout has stated, eventually the light plane will be constructed so that it can land in a field alongside a highway, take on fuel from a regular station, and take off again without being tied up waiting for the arrival of the aviation type of fuel which, at the present time, is not universally available.



■ Fig. 1 – Chart of light plane operating cost (from Terry and Kellogg[1])

The principal objection to the use of automotive fuels seems to come from the possibility of vapor lock and indefinite knock rating. There is no more excuse for vapor lock in an airplane engine than there is in an automobile engine where great strides were made to eliminate vapor lock years ago. In the old days in automobile engineering, very little thought was given to keeping the gasoline in the carburetor at low enough temperatures so vapor lock would not occur. This same condition exists today in many of the light-plane engine installations where fuel temperatures are dangerously near the critical point for automotive fuels as far as vapor lock is concerned.

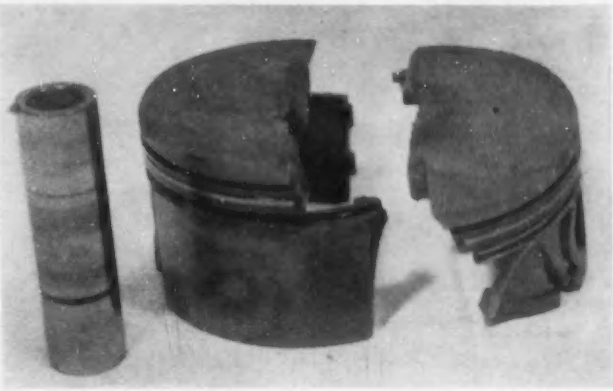
Perhaps the answer would be to change the specifications of automotive fuels to approach the aviation types. However, time only will bring this change about, or until

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 3, 1941.]

[1] See *Aviation*, April, 1941, pp. 34-35, 154, 156, 158: "The Cost of Owning and Operating Small Aircraft," by Cyrl W. Terry and Paul Kellogg.

Their FUEL PROBLEMS

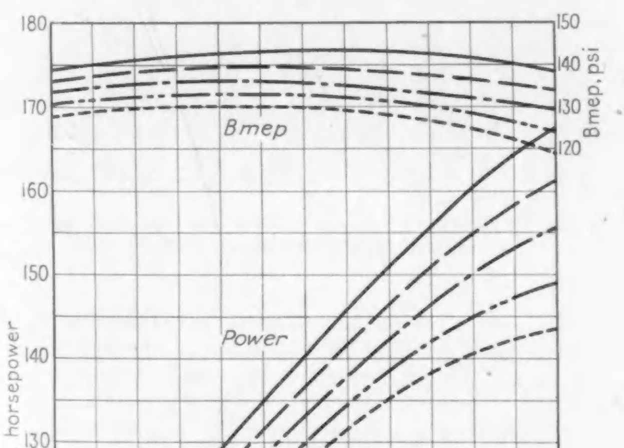
by CARL T. DOMAN
Aircooled Motors Corp.



■ **Fig. 3 – This piston cracked after 30 min operation on 73-octane fuel, instead of 80-octane fuel for which the engine was designed**

we have more and more planes demanding lower-price fuels. While none of the engines manufactured by my company are approved for automobile fuel, I do want to point out that we have conducted satisfactorily four CAA type tests on automotive fuel with absolutely no trouble. A CAA run was made in November, 1940, on our 130-hp 6-cyl opposed engine using 80-octane non-leaded automotive fuel, obtainable in any city.

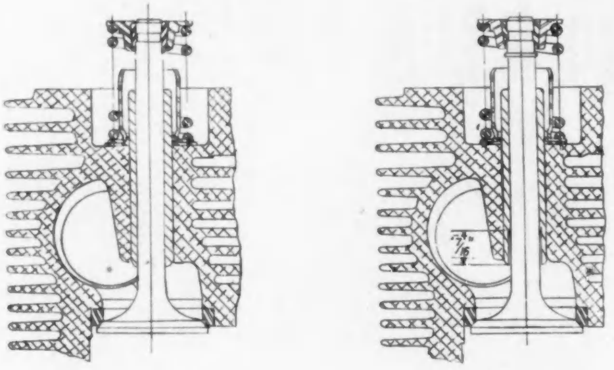
The various methods of rating fuels for knock rating have been most misleading and very much misunderstood. This condition is natural when the customer is confronted with at least five so-called methods of knock-rating, that is, the ASTM method, the CFR method, the Army method, the AFD method (Aviation Fuels Division of the CFR Committee), and the Motor method. In one locality it was found that so-called 80-octane aviation gasoline was actually 73 octane, ASTM method. Certainly all aviation fuels must be rated by the same method or serious trouble will result if the fuel is lower in octane than that specified for the engine. All engines in the light-plane industry are rated for minimum octane requirements by the ASTM method. However, no doubt, shortly the new AFD aviation method will be used.



■ **Fig. 2 – Increase of output with compression ratio at various speeds – Franklin 6AC-298 6-cyl engine; 4¼ x 3½ in. bore and stroke; displacement, 298 cu in.; fuel, as noted for each compression ratio; Marvel carburetors; Eisenmann magnetos**

It is felt that there is a definite need to have a study made to determine the relation between the knock ratings in test engines and actual light aircraft engines, as there are indications that the knock ratings assigned to certain fuels are not correct. We have found in our own laboratory that several different fuels all rated at 80 octane, for instance, will have entirely different knock characteristics in a production engine. Also, it has been found that the knock varies with engine speed, spark advance, and mixture ratio, when different makes of fuel are tested under exactly the same conditions.

It is recognized, of course, that the higher the compression ratio, the higher the output from a given size engine (see Fig. 2). Fuel makers have assured the engine builders that the octane rating is gradually increasing. For that reason the engine makers have designed the engines to accommodate higher-octane fuels. In the past the light-plane engine designer knew that fuels of 65-73-80-87-91-95 and 100 octane were available. Nevertheless, it was felt that 80 octane number was the highest which was practical to use as, in certain localities, fuels above that octane rating were not available. Furthermore, it was found that, in some localities, even 80 octane could not be obtained. Recently, however, the light-plane engine makers have been faced with the problem of the entire rearrangement of the fuel situation, based on the national emergency. This rearrangement has not gone into effect 100% as yet but fuels of 65, either 73 or 80, 91, and 100 octane number will be used eventually; 87 and 95 already have been eliminated; and either 73 or 80 will be eliminated. In the original specifications, 65 and 73 octane fuel contained no lead, with 80 octane fuel containing 0.5 cc maximum of lead. Under the rearrangement of fuel specifications, 73 octane has a maximum of 1 cc of lead; 91 octane has a



■ Fig. 4 – Valve-stem and guide designs

maximum of 3 cc of lead; and 100 octane, 4 cc of lead per gal.

To the light plane engine maker this rearrangement of fuels has resulted in a serious situation where engines were designed and developed to operate on not more than 1 cc of lead per gal. For example, an engine which was developed to operate on 80-octane fuel with not over ½ cc of lead has been found to give trouble shortly when operated on 80 octane fuel with over 1 cc of lead with certain makes of lubricating oil. Furthermore, these light-plane engines have been designed so that it is not possible to reduce the octane rating without running into a multiplicity of other troubles. Thirty min of operation on an engine using 73-octane fuel instead of 80, resulted in a piston cracking (see Fig. 3). At the same time decided corrosion was present in the cylinder-head pocket around the inlet valve. Both of the exhaust valves were badly warped. This result refers to an engine with a maximum bmep output of 146 psi.

Brief mention was made previously of troubles partially attributed to certain makes of lubricating oil. It has been found that, with certain grades of oil and with a lead content of over 1 cc per gal, the lead seems to combine with the so-called "coke" on the valve stems, valve guides, valve faces, and valve seats.

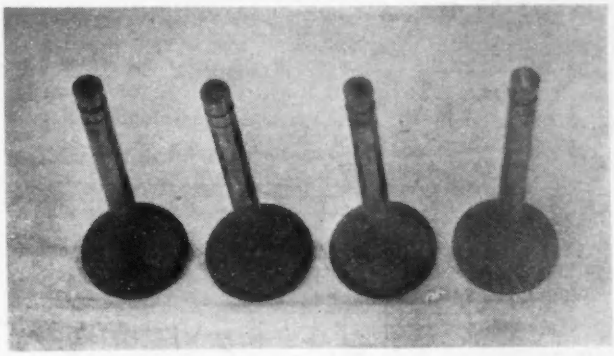
In the attempt to use fuel with higher lead content, still keeping the 7:1 compression ratio requiring 80-octane fuel, various designs of valve guides and valves were tested. Exhaust-valve guides without the relief (see Design *A*, Fig. 4), were tried. After 25 hr of full-throttle operation, the lead had built up in the end of the guide, causing the exhaust valves to drag and, eventually, the ruination of the valves.

The next attempt consisted in using an aluminum-bronze valve guide instead of the all-iron type, with the hope that the increased heat conductivity of the bronze might lower the temperature of the guide and the valve stem so that excessive lead would not deposit. This was found to be of no help over the regular iron guide.

A relief was then put in the lower end of the valve guide 7/16 in. long. This was found to be a decided improvement. However, the greatest benefit seemed to be derived from using the relief in the guide (see Design *B*, Fig. 4). With this change it was found that the engine could operate in the neighborhood of 100 hr full throttle before the lead had deposited sufficiently to cause the valves to drag and burn. With a variable load, the trouble did not develop until approximately 200 hr.

The same engine with iron guides and two-piece exhaust valves, that is, having heads of 21% chrome and 12% nickel, and SAE 3140 stems, would operate indefinitely under full-throttle conditions on a fuel having 0.00 to 1 cc of lead. Investigations proved definitely that the use of a cooler valve would decrease materially and practically eliminate the trouble caused from the higher lead content. However, with light plane engines it is most important that the material bill be kept at a minimum. This is shown in the following table, which represents the major parts expressed in terms of percent of the total cost:

(a) Parts affected directly by the lead content of the fuel	1.3%
(b) Steel parts – forgings, gears, shafts, and so on	19.3%
(c) Aluminum parts	34.5%
(d) Accessories	19.7%
(e) Miscellaneous	25.2%
Total,	100%



■ Fig. 5 – Silcrome X-Be exhaust valves after 100 hr of operation on fuel containing over 1 cc of tetraethyl lead

The use of an all-austenitic valve and bronze valve guides would increase the cost 2.2%, while the use of a salt-cooled valve and bronze guide would increase the cost 6.5%.

One other problem encountered with the use of lead when using a fuel with a lead content of 1 cc per gal was the rapid corrosion of the exhaust valve. In 50 hr with a fuel using 2½ cc of lead per gal, the valves had corroded a maximum of 0.003 in., also the ferrous valve guides had corroded 0.002 in. The use of a bronze guide, of course, eliminates the corrosion as far as the guide is concerned. However, the corrosion on the stem was not eliminated until the all-austenitic valve was used.

Attempts are being made to use a Silcrome X-Be steel, with very promising results thus far. The analysis of this material is as follows:

Carbon	0.60- 0.86
Manganese	0.20- 0.60
Silicon	1.25- 2.75
Chromium	19.00-23.00
Nickel	1.00- 2.00
Phosphorus	0.03 max.
Sulfur	0.03 max.

All indications are that a light plane engine maker without priority must go to a steel with a low nickel content in order to be able to build any engines whatsoever for the private plane. These valves of Silcrome X-Be steel after 100 hr of operation using a fuel of over 1 cc of lead, are shown in Fig. 5.

The foregoing discussion is not intended in any way to discredit the use of additional lead in aviation gasoline, as we all know it to be necessary, especially in these times of national emergency. The idea of this paper is to point out possible means of using a fuel with a lower octane rating, at the same time not lowering the specific output of the engine. There are many approaches to the problem of change in design to permit the use of a lower-octane fuel without sacrificing power. A few of the approaches are as follows:

(a) Increased displacement.

(b) Improvement in installations in airplanes.

(c) Improvement in induction systems, manifolding, and so on.

(d) Changes in cam timing.

(e) Improved cooling of the cylinder and piston.

The problem of increasing the piston displacement is the usual way of increasing the output of an automobile powerplant or, in plain language, the use of a reamer. However, most light plane engines have been designed with no margin for increasing the bore, due to the necessity of keeping the weight to a minimum. Nevertheless, in certain instances, it has been possible to substitute a larger engine with lower compression ratio for a smaller engine with higher compression ratio and still obtain satisfactory performance without seriously increasing the weight. For example, in a certain installation using an 80-hp, 176 cu in. engine, 7:1 compression ratio, weighing 217 lb with starter-generator equipment, it was found possible to substitute an 85-hp, 199 cu in. engine with 6.3:1 compression ratio, requiring only 73-octane fuel. The weight increase was only 8 lb. In many instances, however, the airplane maker absolutely refuses to increase the weight and has insisted that there could be no decrease in power output.

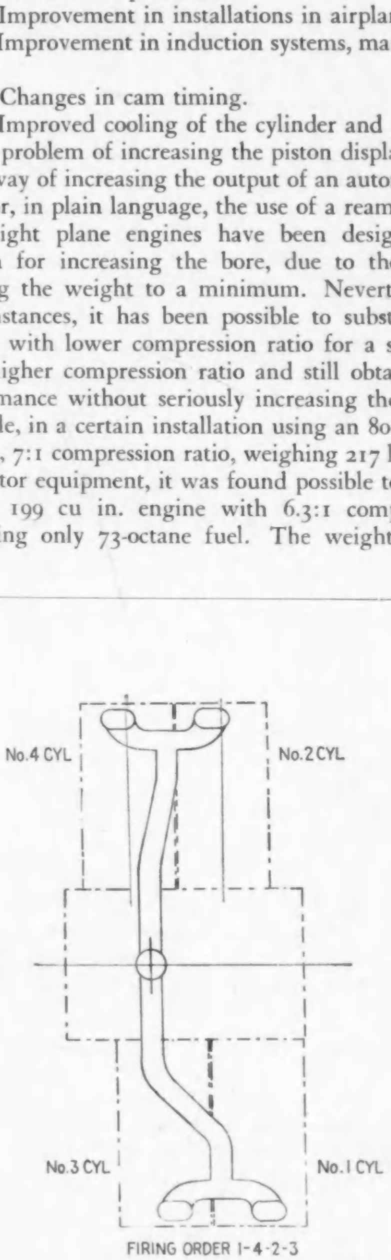
Investigating several production installations which had CAA approval, it has been found that the engines were, in some cases, actually operating 75 F over the maximum temperatures specified for the engine. The use of the higher octane fuel than specified brought the temperatures back to a safe margin. Investigation showed that the air housings or baffles were poorly fitted and did not duplicate those installed for the type test.

This situation, as far as the light plane industry is concerned, is very serious. However, it can, no doubt, be explained by the fact that the majority of the light-plane engineering and production staffs are not powerplant minded and, consequently, do not appreciate the importance of working out the details properly on an installation.

In many instances it is felt that the installation problems are the result of lack of cooperation between the airplane manufacturer and the engine manufacturer. This condition, no doubt, results from the fact that several of the light-plane manufacturers have as many as three different makes of engines in a given model airplane. It is recalled that, in the old days of the automotive industry, the same condition existed, with each powerplant installation a makeshift proposition. Sooner or later each airplane manufacturer for economy's sake alone, will be forced to specify one make of engine for a given airplane. Perhaps that situation is quite far off. Nevertheless, it definitely will come about.

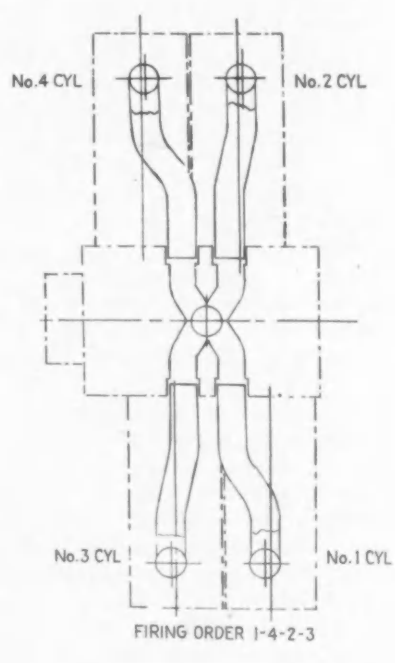
■ Induction System Improved

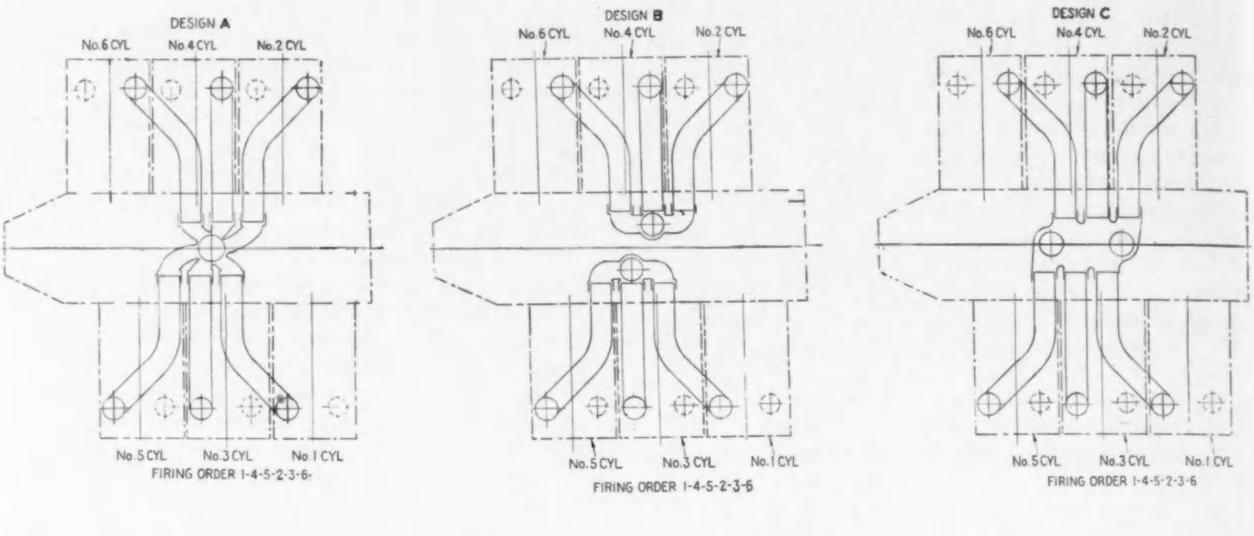
The third approach to the problem of obtaining equal power with low compression ratio and lower-octane fuel is the possible improvement in the induction systems, manifolding, and so on. In work carried on in our organization



■ Fig. 6 (left) – 4-cyl horizontal-opposed manifold design

■ Fig. 7 (right) – Conventional four-port 4-cyl manifold system





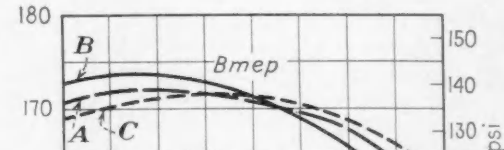
■ **Fig. 8 – Three 6-cyl manifold designs**

it has been found that decided increases in power can be obtained by concentrated effort on the intake manifolding, attempting to tune the system in order to bring about a resonant condition whereby the engine would be naturally boosted. Fig. 6 is a schematic diagram showing the simplest type of manifolding for a 4-cyl opposed engine. This type of manifolding, however, has a decided disadvantage in that with a No. 1 left – No. 4 right – No. 2 right – No. 3 left firing order, No. 1 and No. 2 cylinders have higher pressures due to the fact that they receive the ramming effect from the inertia of the long air column from the carburetor to the inlet port. Fig. 7 shows the conventional four-port system wherein this undesirable feature has been eliminated. Referring to tests with the manifold shown in Fig. 7 it has been found that Nos. 1 and 2 cylinders will at times have compression pressures of 15 psi higher than those in the other cylinders. These same two cylinders also will show a tendency to detonate. The only remedy would, therefore, be to decrease the compression ratio in those two particular cylinders, or to use higher octane fuel.

In designing the manifold for our 6-cyl opposed engine the problem is somewhat simpler, inasmuch as the firing order alternates back and forth across the engine. For example, the normal firing order for a 6-cyl opposed type is No. 1 left – No. 4 right – No. 5 left – No. 2 right – No. 3 left – No. 6 right. In other words, it would be possible to design a manifold with several combinations of carburetors and porting and still obtain good distribution.

Reference to Fig. 8 shows several possibilities for manifold design. Design *A* represents the simplest type and one which has been used to date on the Franklin 6AC-298 engine and the Lycoming o-350.

It has been definitely proved that, in the operating range of a light airplane with a 6-cyl opposed engine, a minimum of 4 to 7% power can be obtained by dual carburetion over the best designed single intake system. It has not been found, however, that the 4-cyl engine can be improved materially by the use of dual carburetion within



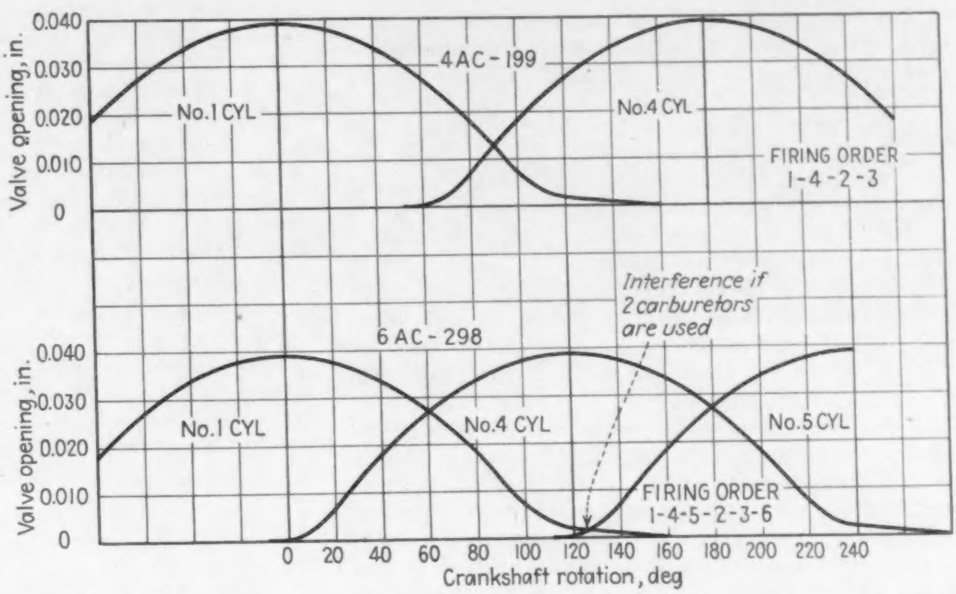
■ **Fig. 10 – Performance of Franklin 6AC-298 6-cyl T-13 engine with three manifolds shown in Fig. 8, A, B, and C**

$4\frac{1}{4} \times 3\frac{1}{2}$ in. bore and stroke; 298 cu in. displacement; compression ratio, 7:1; fuel, Socony Aviation 91; oil, Mobiloil SAE 20; Marvel carburetors; 1 5/16 in. venturi on dual pans, $1\frac{5}{8}$ in. venturi on single pan; mags set 32-32 deg; cam set specified; large valves and long springs

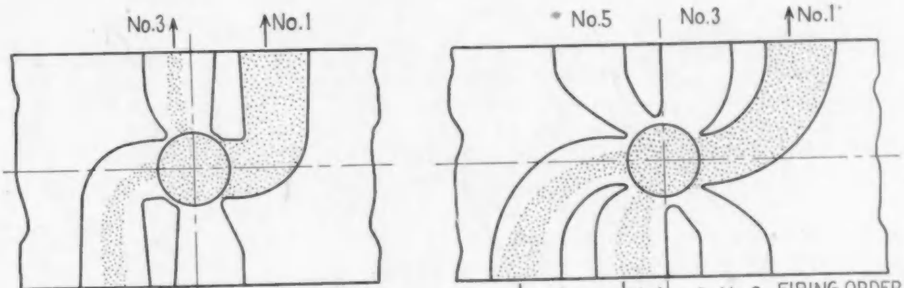
the present operating range. This result is explained by the fact that, on the 4-cyl engine, there is only approximately 60-deg interference between successive cylinders, whereas on the 6-cyl engine there is a possible suction interference of 120 deg (see Figs. 9*A* and 9*B*). Furthermore, the shape of the power curve can be controlled within measure by the particular arrangement or shape of the interior of the manifold. Fig. 10 shows the performance of a 6AC-298 Franklin engine with the conventional manifold shown in Fig. 8*A*. Plotted on this same curve is also the performance obtained from the same engine with the manifolds *B* and *C* of Fig. 8. Examination of these curves is quite interesting, as it shows that it is quite possible to change the performance characteristics of the particular type of application for the engine.

It is noted that, with the manifold arrangement of Fig. 8*B*, the power increases to 2850 rpm, where the single carburetor or the dual carburetion in Fig. 8*C* surpasses it.

For a direct-drive engine with an operating range up to 2850 rpm, the dual carburetor arrangement of Fig. 8*B* would have the greatest advantage. However, if the engine is to be geared where maximum power at take-off is desired, the design in Fig. 8*C* would normally be followed, provided, of course, that the engine at take-off operated from 3400 to 3600 rpm. A sacrifice is taken if the engine is to be used as a direct-drive type where the operating speed would not exceed 2850 rpm. Fig. 11 shows the operation of the 6-cyl engine with a shift type of reduction unit; that is, a gear reduction unit which is in gear for take-off and in direct drive for cruising. From this curve it is seen that, with the shift-type reduction, it is very important that the power at cruising speed is not sacrificed for high take-off power. Otherwise, the engine would be cruised at full throttle the majority of the time. For that reason with this type of reduction unit, a compromise must be made between take-off power and cruising power. On the other hand, if a fixed reduction unit is used with approximately 0.63:1 ratio, it is possible to use an intake system giving maximum power for take-



■ **Fig. 9A – Suction interference study (Timing Diagram 10180) – ratio of interference area of 6-cyl engine with single carburetor compared with 4-cyl engine, 4.1:1**

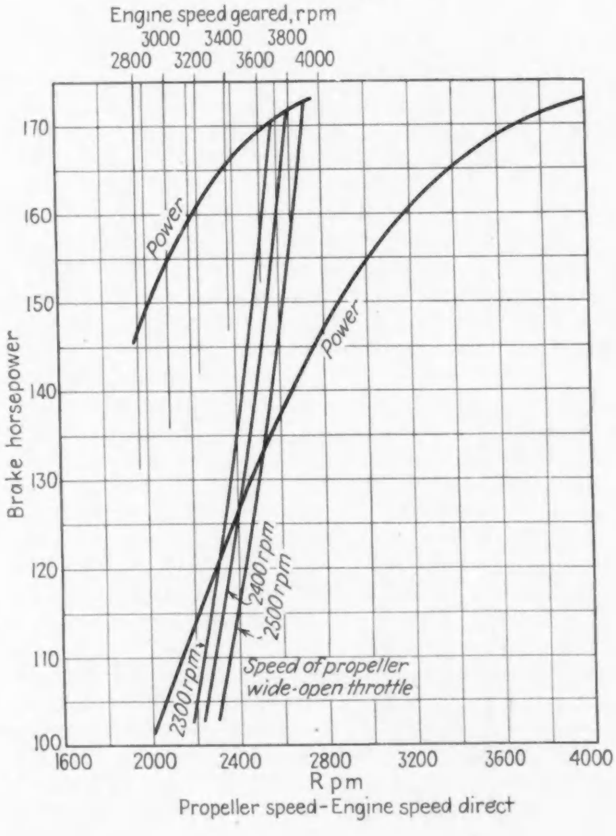


■ **Fig. 9B – Suction flow study – inlet manifold distributing zones**

off (3300-3600 rpm), and up to 90% take-off power for cruising (3000 rpm – see Fig. 11). It is not the purpose of this paper, however, to discuss the relative merits of each type of reduction unit as each has certain distinct advantages.

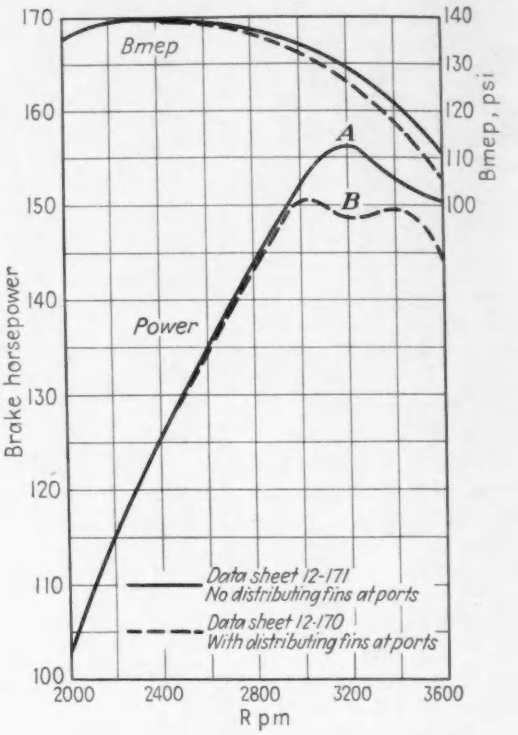
In this study of induction systems, many interesting problems were encountered with very peculiar results. At this time the reasons for some of the peculiar results are not fully explained. However, the work is being continued and shortly it is expected that the answer will be found for these unusual conditions. Reference is made to Fig. 12 which shows a very peculiar break in the power curve at the upper end, which is traceable to the particular shape of the manifold used. While the evidence is not complete, it would appear that the trouble might be caused by the shape of the separators from the individual pipes down to the distributing zone.

In this investigation study was made of the effect of the passage size on output. It was found that it was possible to shift the peak torque over a range of 400 to 600 rpm by the selection of the size of the manifold pipes. The best compromise, however, seemed to be a passage equivalent to 1½ in. inside diameter for the cylinder sizes studied, that is, 4¼-in. bore and 3½-in. stroke. Reference to Fig. 13 shows the results of this study as far as the production engine is concerned. On this curve is plotted the power



■ Fig. 11 – Performance of Franklin 6AC-298 6-cyl T-7 engine with shift type of reduction unit

Bore and stroke, 4¼ x 3½ in.; displacement, 298 cu in.; compression ratio, 7:1; fuel, Socony Special 80; oil, Mobiloil SAE 20; dual Marvel carburetors; 1½-in. venturis; adjustable M.J. magnetos set 30-30 deg; cam set specified



■ Fig. 12 – Performance of Franklin 6AC-298 T-13 6-cyl engine with and without distributing fins at induction-system ports

Bore and stroke, 4¼ x 3½ in.; displacement, 298 cu in.; compression ratio, 7:1; fuel, Socony Aviation 90; oil, Mobiloil SAE 20; dual Marvel carburetors; 1 5/16-in. venturi; adjustable M.J. magnetos set 30-30 deg; cam set specified; large White valves, long springs; L-297 oil pan with partition and ⅜-in. compensating port; 1⅝-in. intake ports

output of the 6AC-298 engine with 7:1 compression ratio using 80-octane fuel. Also on the same curve is shown the output of this same engine with the compression ratio decreased to 6:3 with 73-octane fuel. The final curve shows the output with dual carburetion combined with the lower compression ratio. This curve shows that with dual carburetion the power output in the normal operating range (2550-2600 rpm) at which the engine is approved, is practically the same with dual carburetion and lower compression ratio as with the higher compression ratio and single carburetor.

■ Effect of Cam Timing

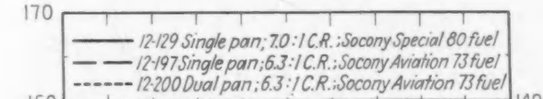
In further attempts to obtain the same output with the lower compression ratio, a study was made to show the effect of cam timing. This work was not successful as it was found that, if the events were changed with the idea of increasing the compression by closing the inlet valve earlier, there was no net increase in output, as it was found that the engine required considerable retarding of the spark advance in order to eliminate incipient detonation. It was also found that, with an earlier closing of the inlet valve, the power decreased at the upper speed range – that is, between 2400 and 2850 rpm – as much as 10%. This result led us to the conclusion that it was wise to continue with the regular valve timing and concentrate on other means for obtaining power with lower compression ratio.

Mention should have been made previously in the paper of the changes in the cylinder design for improved cooling. It has been found that increasing the area of the fins on the head and barrel has helped considerably in permitting the use of higher compression ratio with lower-octane fuels. Fig. 14 on the left shows the original cylinder assembly with 4¼-in. bore and 3½-in. stroke. On the right is shown the present production cylinder which has approximately 17% increase in area on the head and 8% on the barrel, over the original design.

It is not intended that the work discussed in this paper should be considered final, as other experiments are still being carried on for the purpose of actually increasing the output with the lower compression ratio over the engines already approved with the higher compression ratio. The object has been to point out the possibility of an approach to the problem, hoping that other manufacturers of light airplane engines will make similar experiments so that all makers of the smaller engines will benefit.

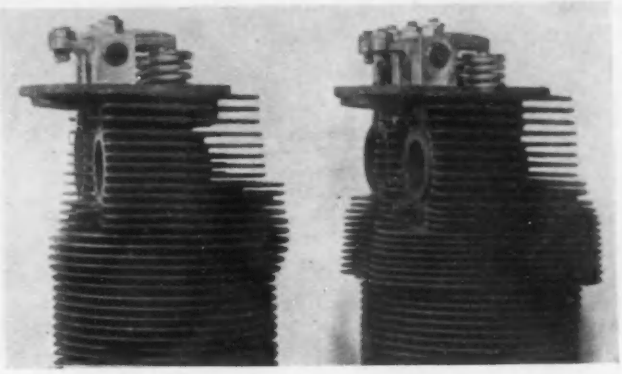
APPENDIX

W. C. Jamouneau, chief engineer, Piper Aircraft Corp. – "We have reached the point now where we would not attach any particular significance to the term light plane, as our aim is to provide a desirable airplane for the private owner trade and, in this respect, we believe that the future conception of the light plane will be of the family type, with a low first cost and with economical operating characteristics.



■ **Fig. 13 – Effect of various combinations of pan, compression ratio, and fuel on performance of Franklin 6AC-298, T-13, 400122 6-cyl engine**

Bore and stroke, 4¼ x 3½ in.; displacement, 298 cu in.; compression ratio and fuel as noted; oil, Mobiloil SAE 20; Marvel carburetors; single, 1⅜-in. venturi; dual, 1 5/16 in. venturi; magnetos set – T-13, 30-30 deg; 400122, 28-28 deg; cam set specified



■ **Fig. 14 – Experimental (left) and production 4¼ x 3½-in. cylinder showing increased finning area in production design**

"In our estimation the limits bounding a light plane would be a maximum of 1500 lb gross weight and 125 hp."

James G. Rising, chief engineer, Luscombe Airplane Corp. – "It would be difficult to set up a light classification that would agree with popular conception, since some people judge according to price, some according to engine, horsepower, and still others according to flying characteristics. Probably a 1300-lb upper limit would be more reasonable to most people."

R. W. Rummel, chief engineer, Rearwin Aircraft & Engines, Inc. – "For a definition in terms of horsepower and standard or gross weight as requested, although I don't believe this to be the proper criterion applicable to all considerations, I would stand by that given in my paper 'Maintenance Development in Light Aircraft' with the possible exception of increasing the power limit to 125 hp due to recent engine developments."

William R. Skinner, chief engineer, Porterfield Aircraft Corp. – "In my opinion there are no true light airplanes in production in this country at the present time. I feel that the true light airplane went out with the increase of horsepower from 40 up.

"Of course, you understand that the foregoing is merely my personal definition of a true light airplane. As far as popular opinion is concerned, I agree with you that it will be necessary to move the power requirements to as much as 150 hp in the smaller privately operated aircraft. However, I think it will be some time, perhaps as much as five years, before this evolution completes itself."

A. P. Fontaine, chief engineer, Stinson Aircraft Division, Vultee Aircraft, Inc. – "I feel that, rather than stretch the classification of light airplanes to include all of these higher powered airplanes, it would be desirable to establish a medium class airplane. I feel that 65 hp with a gross weight of 1300 to 1400 lb should still be the upper limit of truly light airplanes. I feel that an airplane such as our 1940 '105' with 90 hp and over 1600 lb is not in a light plane class."

James A. Weagle, chief engineer, Aeronautical Corp. of America. – "In line with your thought that a light airplane would be powered with 100 hp or more, we believe this to be entirely possible providing the wing loading is kept below 10 lb per sq ft. Of course, in an airplane of this type, I believe that the power loading would enter into the matter the same as the wing loading more or less designates a present-day light airplane. In my opinion, I would set the power loading of a light airplane between 17 and 20 lb per hp, and the wing loading less than 10 lb per sq ft."

Changes Occurring in OILS

THE problem of lubricating automotive engines is an active one which is far from completely solved even though each succeeding year shows marked improvement in the quality of the average lubricant. The engine manufacturers' desire or necessity of improving engine design will no doubt prevent complete solution of all lubricating problems. Before lubricants are developed to withstand all present engine punishment, more engine punishment for the oil will be provided.

In a broad general consideration of lubricating oils and engines, there are two questions of almost equal interest. One concerns the changes in lubricating oil while in service in an engine. The other concerns the changes which occur in an engine in service with a lubricating oil. From a theoretical viewpoint, there should probably be no change in either the lubricating oil or in the engine during use but, from a practical viewpoint, we are a long way from any such perfect system. It may be assumed, however, that any change which occurs either in the lubricating oil or in the engine during use is objectionable.

There are numerous factors which determine the nature and degree of the changes in both the engine and the oil during use. Some of the known factors are physical and chemical nature of the lubricating oil; design and condition of the engine; conditions of service, such as speed, load, duration, ventilation, temperature, and extraneous material from fuel, cooling system, combustion zone, air, or other sources. Each of these factors and various combinations may affect different lubricating oil-engine systems differently.

Accordingly, the study and rating of oils and engines must include a study of many combinations of the more severe of all of these factors. Reliable information cannot be obtained by one piece of laboratory equipment, by a single type of engine, or by a single set of service conditions. To rate accurately lubricating oils or engines, a large number of tests on several pieces of equipment operated under a number of different sets of operating conditions is required. Each distinct test will produce a large number of types and degrees of change in both the oil and the test equipment, which must be isolated, evaluated, and recorded for comparison. Such a test program faithfully carried out will produce a volume of data so large that conclusions and predictions are impossible without tabulating, correlating, and digesting all the data.

In outline, data are presented to show the changes which occurred in the lubricating oils and the engines during use in a comprehensive study which culminated in the road

THE paper, "Review of Temperatures, Road-Test Engines,"[1] presented at the Semi-Annual Meeting of the Society, June 5, 1941, by B. E. Sibley, gave operating temperature data on a fleet of 16 road-test units operated in southern Arizona during August, 1940.

This paper deals with the changes occurring in the five oils used in that test work and with the nature and amount of deposits left in the various engines by the oils.

Results of oxygen-absorption tests and modified Underwood tests of the five oils are given and show the general behavior characteristics that might be expected of the oils in service. Dynamometer stand tests of the same oils in two each of three makes of large production passenger-car engines are described and the oils are rated in order of descending merit, as measured by progressive oil change and final engine condition.

Changes occurring in both oil and engine condition during extensive road tests of these oils are then shown and compared with chemical laboratory and engine-stand test results.

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[This paper was presented at the National Fuels and Lubricants Meeting of the Society, Tulsa, Okla., Oct. 24, 1941.]

[1] See *National Petroleum News*, Vol. 33, No. 34, Aug. 20, 1941, pp. 267-270: "Review of Temperatures, Road-Test Engines," by B. E. Sibley and H. C. Baldwin.

[2] See *The Journal of Industrial and Engineering Chemistry*, Vol. 14, April 1922, pp. 269-278: "A New Method of Color Measurement for Oils," by L. W. Parsons and R. E. Wilson.

[3] "Thialkene Inhibitor," U. S. registered trademark, Continental Oil Co., U.S. Patent 2,218,132, Oct. 15, 1940.

[4] U.S. Patent 1,944,941, Continental Oil Co. See Proceedings of the Sixteenth Annual Meeting of the API, November, 1935, Los Angeles, Calif.: "Practical Selection of Improved Lubricants," by L. L. Davis, Bert H. Lincoln, and B. E. Sibley.

[5] U. S. Patents 2,113,810; 2,113,811; 2,172,285; 2,186,646; and 2,204,538, Continental Oil Co.

[6] See *The Journal of Industrial and Engineering Chemistry*, Vol. 33, March, 1941, pp. 339-350: "Oxidation of Petroleum Lubricants," by L. L. Davis, B. H. Lincoln, G. D. Byrkit, and W. A. Jones.

and ENGINES from USE

by FRANK A. SUESS, W. A. JONES, H. C. BALDWIN, and BERT H. LINCOLN

Continental Oil Co.

tests mentioned by Sibley et al.[1] Prior to these road tests, a very extensive study was made of 12 commercial oils marketed as premium-grade oils by the predominant oil companies, and of 14 experimental or development oils. Of these 26 oils, 3 of the commercial oils and 2 of the experimental oils were found to display outstanding merit. These oils were therefore selected for extensive road tests, and this paper includes a discussion of the service changes in these oils and engines under the following classifications:

I. Lubricants and Equipment Used.

II. Chemical Laboratory and Mechanical Laboratory Investgations.

III. Road Tests.

The data support the observations and conclusions presented and permit others to evaluate the data for different or additional conclusions.

I. Lubricants and Equipment

Since these investigations were made on certain lubricating oils and certain investigational equipment, it is necessary to define each clearly.

Lubricants – Data from the investigation of five lubricating oils manufactured by some of the largest and most reliable oil companies have been tabulated and correlated for this report. Not all of the five oils were tested on all of the groups of road-test units.

The physical and chemical characteristics of the five oils are listed in Table 1. Oil II is a solvent-treated, Midcontinent paraffin base oil blended with two addition agents. One of these addition agents is "Thialkene Inhibitor,"[3] a sulfur-type antioxidant and corrosion inhibitor. The other is "Methyl di-Chlor Stearate,"[4] added to increase oiliness and load-carrying capacity. Oil I is identical with Oil II with the exception that a different sulfur-type antioxidant[5] and corrosion inhibitor is used. The other three oils selected from the results of the preliminary chemical and mechanical laboratory investigations represent the top-quality lubricating oil of three of the leading manufacturers. Test conditions were designed to represent the most severe conditions likely to be encountered in service and were sufficiently severe to reveal differences in superior grades of oils.

Equipment – The equipment used in this series of tests includes the oxygen-absorption apparatus and the modified Underwood apparatus for chemical laboratory testing. Two each of three makes of engines, groups F, G, and H, mounted on stands and connected to calibrated fans, were used for mechanical laboratory testing. Four groups of automotive equipment were used in high-speed, high-temperature road tests. Each group consisted of four identical units, and these groups will be referred to as A, B, C, and D.

Chemical Laboratory – The oxygen-absorption test described by Davis et al[6] indicates the tendency of a particular oil to absorb oxygen and its sensitiveness to the catalytic effect of metal in organic combination in accelerating oxygen absorption.

The modified Underwood[6] test gives information regard-

Table 1 – Physical Characteristics of Oils Tested

	Oil I SAE 20	Oil II SAE 20	Oil III SAE 20	Oil IV SAE 20	Oil V SAE 20	Oil I SAE 30	Oil II SAE 30	Oil III SAE 30	Oil IV SAE 30	Oil V SAE 30
Gravity	30.6	30.6	31.3	29.3	29.1	30.6	30.7	28.9	28.7	29.5
Flash, F	450	450	415	435	430	480	480	450	450	445
Fire, F	515	515	470	500	490	550	550	510	510	510
Viscosity at 100 F, centistokes	70.8	70.8	66.45	72.95	74.9	93.3	93.8	121.8	109.1	91.8
Viscosity at 100 F, SSU	327	327	307	337	346	431	433	562	504	424
Viscosity at 210 F, centistokes	8.71	8.71	8.77	8.39	8.77	10.42	10.48	12.42	10.80	10.38
Viscosity at 210 F, SSU	54.8	54.8	55	53.7	55	60.7	60.9	67.9	62.0	60.5
Viscosity Index	104	104	112	91.5	100	100	102	101	89	103
Pour Point, F	5	5	−20	0	−5	10	10	−10	0	20
Color, ASTM	3−	3−	5+	1½+	4½+	2½	3	5	2+	5−
Color, true color units*	16	16	44	4	37	13	18	42	9	40
Steam Emulsion Number	90	90	960	90	120	90	90	420	150	150
Herschel Demulsibility	1200	1200	20	600	1620	1200	1620	270	1320	1620
Neutralization Number	0.03	0.025	0.05	0.03	0.05	0.03	0.015	0.06	0.04	0.05
Saponification Number	2.50	1.84	0.30	0.45	0.20	2.46	1.34	0.31	0.37	0.21
Conradson Carbon Residue, %	0.01	0.01	0.28	0.04	0.33	0.03	0.04	0.37	0.06	0.21
Sulfur, %	0.38	0.38	0.10	0.21	0.18	0.18	0.18	0.23	0.21	0.08
Chlorine, %	0.068	0.070	0.00	0.00	0.00	0.068	0.073	0.00	0.00	0.00
Phosphorus, %	0.00	0.00	0.010	0.012	0.010	0.00	0.00	0.008	0.012	0.00

* In the examination of oils, the color is reported in true color units determined by the procedure of Parsons and Wilson [2]. The use of this color scale also permits comparison of colors of used oils which are far darker than the range of the conventional color scales.

ing the tendency of an oil to form products corrosive to metals. This test equipment differs from an automotive engine in that extrinsic soluble or reactive catalysts, such as lead from the combustion chamber, do not contaminate the oil during the test. Some information is given by the modified Underwood test regarding the predominating types of deterioration products which the oil will have a tendency to develop in automotive service.

Mechanical Laboratory – Engine Groups F, G, H – Stand Tests – The engines of groups F, G, and H used in the mechanical laboratory were selected to give the maximum amount of information concerning the oils tested and their effect on engine condition and performance. Accordingly, the engines were selected to include the greatest range of engine materials and engine design consistent with the limits of a reasonable number of test engines. It was found that three makes of the larger production passenger-car engines included substantially all of the more important factors normally expected to cause variation in oil-engine systems in service. Two engines of each make were operated simultaneously with each oil to provide check runs to assure the accuracy of the data. The variables in engine design included:

1. Pressure and pressure-splash lubricating systems.
2. Low-pressure and high-pressure lubricating systems.
3. Overhead and L-head designs.
4. Trunk and constant clearance or T-slot pistons.
5. Tin-plated cast-iron and aluminum-alloy pistons.
6. All rings above the piston pin and rings both below and above piston pin.
7. Rolled-bronze and aluminum-bronze piston-pin bushings.
8. Crankcase ventilation rates from 26 to 225 cu ft per hr at 60-65 mph speed range, as specified by the respective manufacturers.

In the operation of these three groups of engines, a uniform high-grade fuel with constant tetraethyl-lead content was used. The loads were maintained constant by fans attached to each engine. The fan-loading device was calibrated to absorb the horsepower required, according to the engine manufacturer's specifications, to drive a sedan of that make at 60 mph. Controlled forced-draft crankcase ventilation was maintained at the rate specified by the manufacturer of each engine.

Entire engine interiors were cleaned thoroughly and inspected after each run. Before each test, piston rings and bearings were replaced, valves were ground, and the engine brought to new standard condition. A standard oil was used for engine "balancing runs" before the start of the series of tests and periodically during the work to avoid the effects of engine condition "drifting" from standard. The engines usually required only from 2 to 5 qt of additional oil in 5000 miles, thus indicating minimum oil consumption and "sweetening."

Operating conditions were as follows:

Engine	Group F	Group G	Group H
Load, hp	26.5	29	23.5
Speed, rpm	3040	3150	2960
Speed, mph	60	60	60
Oil gallery temperature, F	280	280	280
Water outlet temperature, F	185	185	185
Crankcase ventilation, cu ft per hr	32	26	225
Oil pressure, psi	11-15	30-35	30-35
Crankcase drains during test	None	None	None
Duration of test, miles nonstop	5000	5000	5000

Road-Test Equipment

Engine Groups A, B, C, D – The road-test equipment consisted of four each of three different high-production, popular-priced makes of 1940 coupes, designated as Groups A, B, and C, and four 1940 trucks equipped with bus axles of 5.22:1 ratio, designated as Group D. All units complied with stock engine specifications. A uniform "regular-grade" fuel with constant tetraethyl-lead content was used in all units throughout the test.

The road-testing activity occurred during August, 1940, and was conducted over the highly improved roads traversing the desert-like country of a part of southern Arizona, where high temperatures prevail at that time of year and where traffic conditions permitted continued high-speed operations.

Each day the engines were given a 15-min "warm-up" as described by Sibley[1] before going on the road. All the test units were operated under a rigid schedule of actual speed, uniformity of speed, and spacing between units.

Two shifts of drivers were used each day. This arrangement required 24 drivers for car Groups A, B, and C, and 8 drivers for truck Group D. The positions of drivers remained constant, and the position of test units moved one position each shift change. By this method, each car driver would have the same car every twelfth day. In the same way, each truck driver would have the same truck every fourth day.

Group B is the same make of engine as Group F used in the mechanical-laboratory engine tests. Group A is very similar in engine design and characteristics to Group H of the mechanical-laboratory engines.

In addition to the variables of design and materials provided by the mechanical-laboratory engines, the road-test units included:

1. Engines equipped with three different makes of oil filters supplied as standard engine equipment, and engines equipped with no filters.
2. Cast-iron and aluminum cylinder heads.
3. Tin-babbitt, copper-lead, and cadmium-alloy bearings.
4. In-line and V-type engines.
5. Mechanical and zero-lash hydraulic valve-lifter designs.

Operating conditions were as follows:

Unit	Group A	Group B	Group C	Group D
Payload, lb	Driver plus 250 lb	Driver plus 250 lb	Driver plus 250 lb	Driver plus 13,200 lb
Average road speed, mph	62	62	62	52
Average atmospheric temperature, F	96	96	96	96
Average oil gallery temperature, F	198.1	198.3	192.0	157.9
Water outlet temperature, F	174.0	171.4	170.5	165.0
Miles traveled per day	578	578	578	432
Break-in miles	2,500	2,500	2,500	2,500
Duration of test	10,080	10,080	10,080	10,086
Miles between drains	2,500	2,500	2,500	2,500

II. Laboratory Investigations

Various engine designs tend to accentuate different types of changes in lubricating oil, and various oils tend to accentuate different types of conditions in an engine. The lubricating oils and engines used in these investigations were developed for general use. To determine the probable changes which might occur in both the oils and engines

during a road test program, investigations were made in special equipment under conditions accelerated to induce these possible changes in the lubricating oils and engine conditions. Determining the nature and degree of the engine and oil changes by chemical laboratory and mechanical laboratory investigations was of untold value in interpreting data obtained from the road tests. Moreover, the confirmation of road-test data by chemical and mechanical laboratory data justifies more positive conclusions from road tests.

Some of the characteristics of these oils and these engines determined by chemical and mechanical laboratory investigations include:

(a) Effect of oxygen and metallic accelerators on the oxidation of the oil.

(b) Effect of time, temperature, air, and certain accelerators on corrosion, viscosity increase, asphaltene formation, sludge, and resin formation.

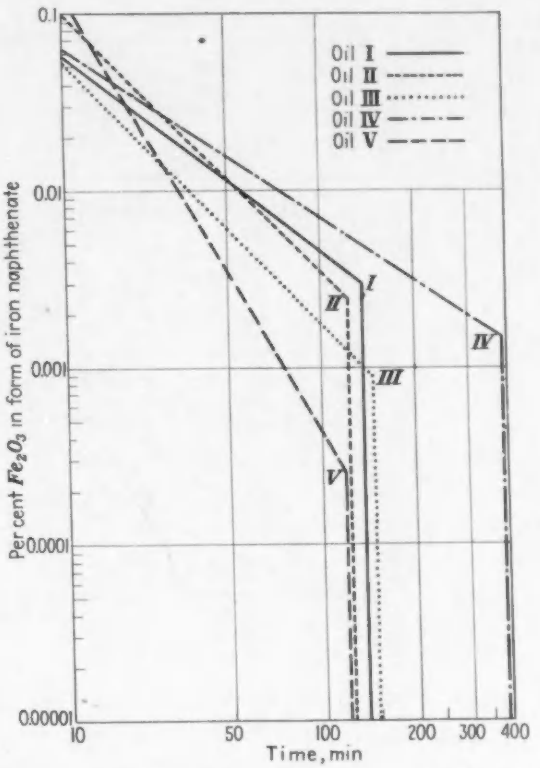
(c) Effect of all of the conditions of internal-combustion engine operation on the nature of engine dirt, that is whether soft and oil suspendable or whether of a hard granular nature which may cause an increased compression ratio with or without abrasion, and nature and degree of engine lacquer.

Some of the products of oil change may be more objectionable in certain engine designs and less objectionable in others. For this reason, to be significant, road testing must include the use of engines which accentuate each of the types of oil and engine changes found objectionable in practical lubrication. These road tests must be accompanied by laboratory tests designed to accelerate these tendencies to change; therefore, while road tests are indispensable, they are no more infallible than laboratory tests but are the closest approach to service problems.

It has been necessary to condense very greatly the mass of data secured in this testing program both because of lack of space and because of the attempt to present the results in comprehensible form. All of the five oils were tested in the chemical laboratory and on all of the groups of mechanical-laboratory engines, but the number of road-test units involved prevented testing all the oils in all groups of road test units.

The percentage of change in each oil on each test has been calculated, calling Oil I the reference oil and rating its changes as 100%. In all comparisons, a higher percentage represents a larger amount of change in the oil or engine condition and a lower percentage, a lesser amount.

The average percentage change based on Oil I for each oil and type of testing equipment is simply an unweighted average of all the types of change noted. It is realized that differences of opinion exist as to the relative importance of the various types of change in both oil and engine condition in service, and no attempt has been made to give each type the correct weight. Accordingly, the averages shown will not necessarily represent the ratings which everyone would agree to as a proper evaluation of the oil. The data show that changes occurred, and it is assumed that all change is equally undesirable.



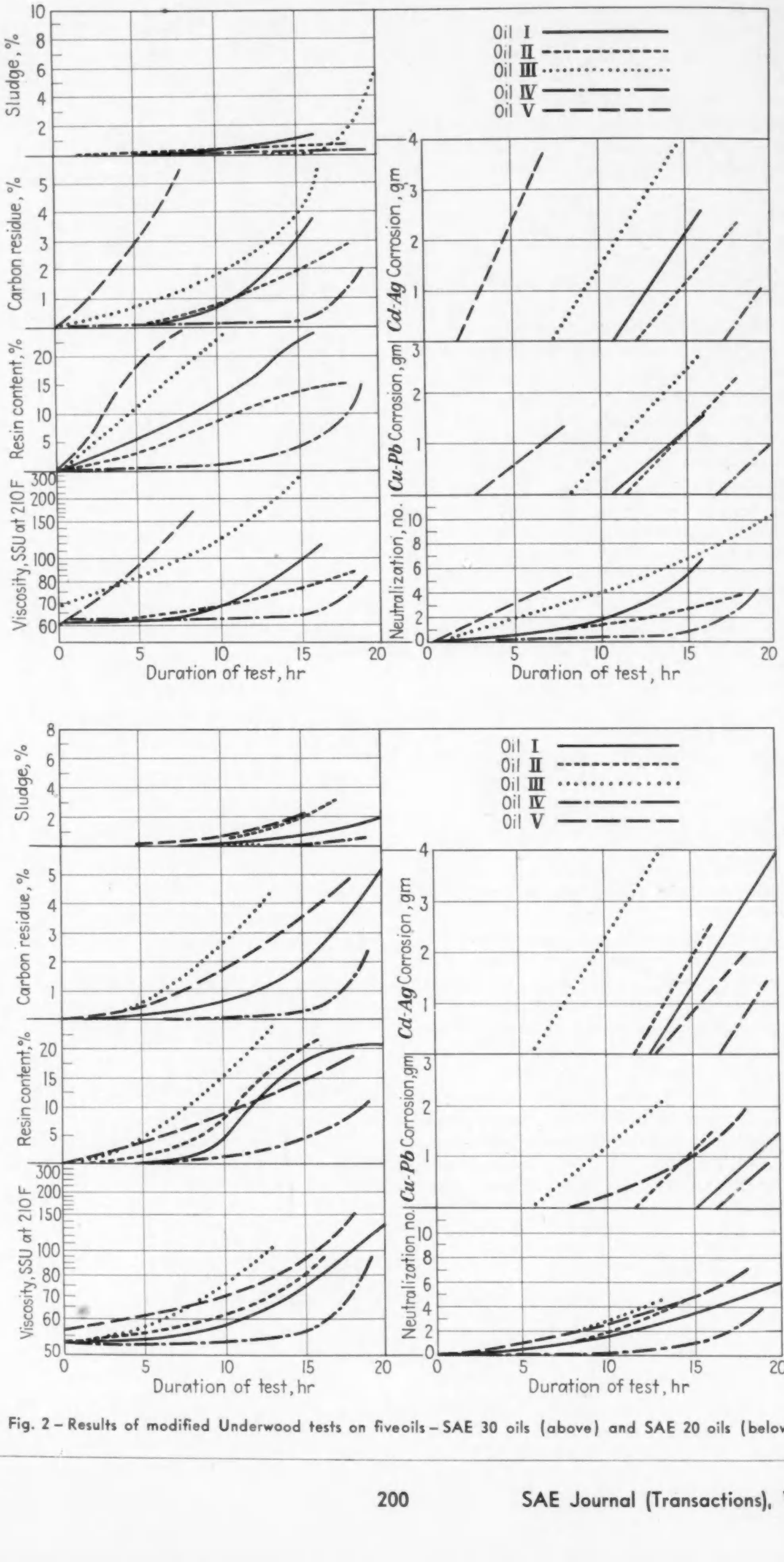
■ Fig. 1 – Results of oxygen-absorption test on five oils

Oxygen Absorption – The results of the oxygen-absorption test described by Davis et al[6] are shown in Table 2 and Fig. 1. The "stability" shown is the time in minutes required for the oil, without metal-containing catalyst, to absorb sufficient oxygen to cause a drop in pressure of 60 mm. The "iron tolerance" shown is the amount of iron naphthenate in "per cent Fe_2O_3" required by each oil to produce a perceptible acceleration in the rate of oxygen absorption. The reciprocal of these percentages of iron oxide measures the sensitivity of the oils to catalytic oxidation. This tendency is shown in Table 2 in percentage for each of the five oils with Oil I arbitrarily given a rating of "100%."

The oxygen-absorption test gives no information of the type of oil deterioration products that might be expected in engine service. It indicates only the relative tendency of oils to form oxidation products which may remain in solution or in suspension in the oil or be deposited on engine surfaces. From this table, the amount of oxidation

Table 2 – Oxygen Absorption Test – SAE 20 Oils

	Oil I	Comparative Percentage of Reciprocals	Oil II	Comparative Percentage of Reciprocals	Oil III	Comparative Percentage of Reciprocals	Oil IV	Comparative Percentage of Reciprocals	Oil V	Comparative Percentage of Reciprocals
Stability, min	138	100	121	114	148	93	389	35	120	115
Iron Tolerance, % Fe_2O_3	0.0030	100	0.0025	120	0.0009	333	0.0015	200	0.00025	1200



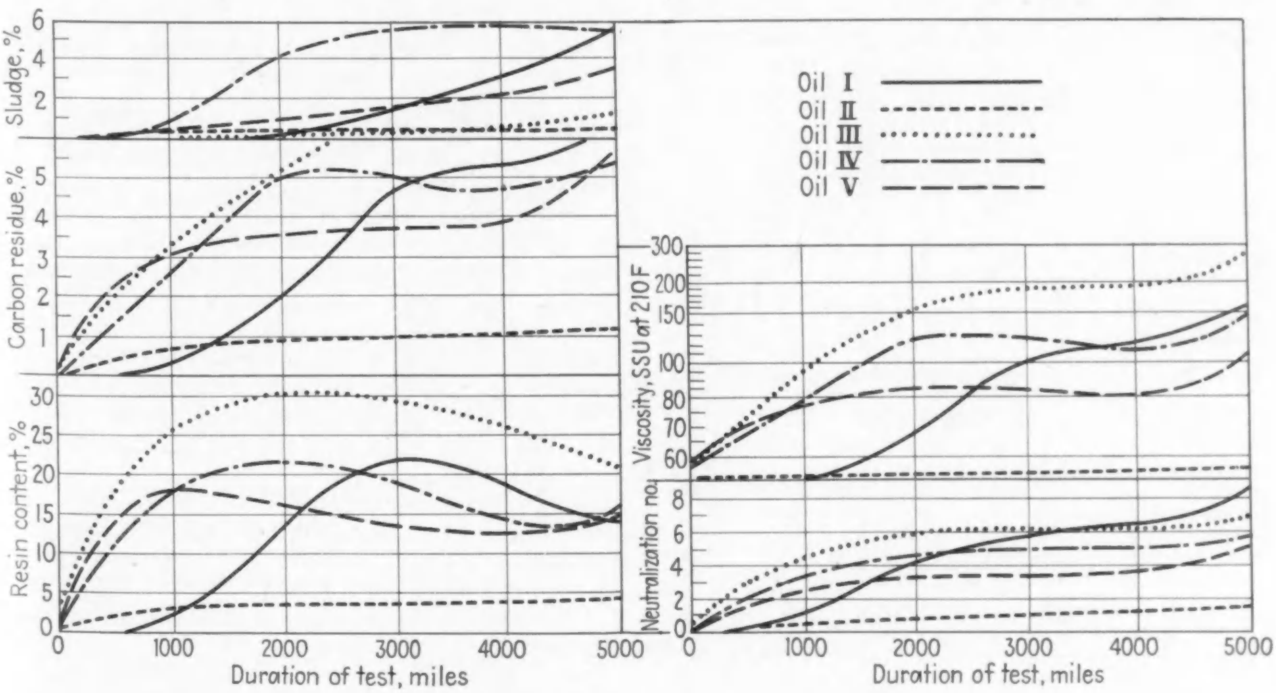
■ Fig. 2 – Results of modified Underwood tests on fiveoils – SAE 30 oils (above) and SAE 20 oils (below)

Table 3 – Modified Underwood Tests

	Oil I		Oil II		Oil III		Oil IV		Oil V	
	Hr	Comparative Percentage of Reciprocals	Hr	Comparative Percentage of Reciprocals	Hr	Comparative Percentage of Reciprocals	Hr	Comparative Percentage of Reciprocals	Hr	Comparative Percentage of Reciprocals
SAE 30										
Hours before bearing corrosion:										
Copper-lead	11	100	12	91	9	122	17	65	3	366
Cadmium-silver	11	100	12	91	9	122	17	65	2	550
Hours to develop:										
Neutralization No. of 3.0	12	100	17	71	8	150	18	67	5	240
Sludge of 1.0%	18	100	20	91	18	100	24	76	24	76
Viscosity increase of 25%	12	100	14	86	6	200	18	67	4	300
Conradson carbon residue of 2.0%	13	100	13	100	10	130	19	69	4	322
Resin of 10%	9	100	10	90	5	180	18	50	2	450
Average		100		89		173		67		329
SAE 20										
Hours before bearing corrosion:										
Copper-lead	15	100	11	136	5	298	16	93	8	187
Cadmium-silver	12	100	12	100	6	200	16	77	13	93
Hours to develop:										
Neutralization No. of 3.0	14	100	12	117	10	141	18	77	11	128
Sludge of 1.0%	16	100	12	134	24	68	20	81	12	134
Viscosity increase o 125%	14	100	13	108	8	176	18	77	10	141
Conradson carbon residue of 2.0%	15	100	14	106	9	166	19	79	11	136
Resin of 10%	12	100	11	109	8	151	18	67	11	109
Average		100		111		180		79		133

products formed in Oil II and Oil I would be expected to be quite similar under all conditions of service. Oil III should form less oxidation products than Oils I and II in the absence of oxidation accelerators, such as dissolved metal compounds. It is, however, indicated to be more sensitive than Oil I to the catalytic effect of metal compounds which are always encountered in automotive service.

Oil IV is indicated to possess a high inherent stability to straight oxidation but to be more affected by catalysts than is Oil I. Of the entire group, Oil V is the least stable and most sensitive to catalysts and would be expected to permit the formation of the largest amounts of oxidation products. This test rates the sensitivity of the oils to catalytic oxidation in the order of descending merit as: 1, 2, 4, 3, and 5.



■ Fig. 3 – Results of tests of five SAE 20 oils in Engine F-1

Table 4 – Progress of Formation of Products of Change in Oil I in Mechanical Laboratory Test Engines

Group F – Engine No. 1 – Oil I

Miles	1000	2000	3000	4000	5000
Neutralization No.	0.5	4.3	6.0	6.4	8.5
Sludge, %	0.15	0.20	1.82	3.28	5.54
Chloroform Insoluble, %	0.08	0.14	1.66	3.01	4.83
Viscosity at 210 F, SSU	54.7	66.4	106.2	116.2	161.0
Carbon Residue, %	0.41	1.91	4.97	4.9	6.4
					*F 5.9
Resin, %	2.18	13.80	24.06	18.31	16.82
Iron as Fe, %	....	....	....	....	0.037
					F 0.032
True Color	41	376	2680	3640	5700

Group F – Engine No. 2 – Oil I

Miles	1000	2000	3000	4000	5000
Neutralization No.	0.7	6.0	4.2	6.5	4.0
Sludge, %	0.31	1.11	1.38	2.85	2.95
Chloroform Insoluble, %	0.23	0.96	1.04	2.56	2.35
Viscosity at 210 F, SSU	56.4	87.8	77.5	116.0	88.5
Carbon Residue, %	0.53	3.42	2.27	4.09	2.8
					F 1.05
Resin, %	2.77	20.81	16.21	18.84	13.02
Iron as Fe, %	....	....	....	....	0.019
					F 0.007
True Color	70	846	959	1260	780

Group G – Engine No. 1 – Oil I

Miles	1000	2000	3000	4000	5000
Neutralization No.	2.2	6.5	6.5	7.0	4.5
Sludge, %	0.50	1.45	1.86	1.60	0.40
Chloroform Insoluble, %	0.43	1.24	0.21	0.48	0.06
Viscosity at 210 F, SSU	58.0	98.8	104.8	109.3	78.5
Carbon Residue, %	0.82	3.71	2.9	2.6	1.6
					F 1.5
Resin, %	6.81	22.54	22.82	22.31	16.80
Iron as Fe, %	....	....	....	....	0.007
					F 0.006
True Color	186	931	884	799	583

Group G – Engine No. 2 – Oil I

Miles	1000	2000	3000	4000	5000
Neutralization No.	2.7	7.0	5.8	5.0	5.0
Sludge, %	0.35	1.12	1.60	0.99	0.53
Chloroform Insoluble, %	0.27	1.01	0.77	0.77	0.41
Viscosity at 210 F, SSU	59.6	101.1	104.1	93.2	83.2
Carbon Residue, %	0.99	3.91	2.86	2.42	2.11
					F 1.79
Resin, %	9.32	24.26	18.69	19.31	16.21
Iron as Fe, %	....	....	....	....	0.008
					F 0.006
True Color	256	940	1730	667	639

Group H – Engine No. 1 – Oil I

Miles	1000	2000	3000	4000	5000
Neutralization No.	3.5	6.6	6.5	6.5	8.4
Sludge, %	0.40	1.06	2.77	3.42	5.41
Chloroform Insoluble, %	0.34	0.97	2.62	3.20	5.16
Viscosity at 210 F, SSU	61.4	89.3	115.1	104.1	144.8
Carbon Residue, %	1.30	3.64	4.67	3.69	4.87
					F 2.18
Resin, %	7.31	20.44	20.49	15.58	18.81
Iron as Fe, %	....	....	....	....	0.053
					F 0.020
True Color	179	1205	3280	611	855

Group H – Engine No. 2 – Oil I

Miles	1000	2000	3000	4000	5000
Neutralization No.	1.2	6.1	4.5	6.0	5.5
Sludge, %	0.26	0.29	0.93	1.01	1.20
Chloroform Insoluble, %	0.21	0.25	0.56	0.84	0.61
Viscosity at 210 F, SSU	56.7	76.5	78.0	100.4	96.7
Carbon Residue, %	0.67	2.99	2.18	2.69	2.55
					F 2.39
Resin, %	5.13	16.11	16.82	19.93	22.36
Iron as Fe, %	....	....	....	....	0.011
					F 0.010
True Color	80	667	688	874	752

* F – Sludge-free oil.

Modified Underwood – The results of tests in an all-steel modified Underwood apparatus[7] at 350 F using 15 sq in. of sheet lead in the heater chamber are shown in Table 3 and Fig. 2.

The results of the modified Underwood test are shown as time in hours required to produce an arbitrary amount of each of the products of oil change. The reciprocals of these time periods are a measure of the rate of formation of each of these products. These rates of formation of each of these products by the five oils are shown in percentages.

Both cadmium-silver and copper-lead bearings were used, and this test indicates fairly well the relative corrosiveness of oils in engines using these bearings. No oil-soluble metallic catalysts are added to the oil in this test. A time period is required for the formation of oxidation products. The length of this time period is fixed by the stability of the oil to oxidation, but not by its sensitivity to the effect of added catalysts. When corrosive oxidation products are formed, they will corrode the bearing metals, iron of the Underwood parts, and the lead sheet, thus forming soluble metal-containing compounds. All of these metal-containing compounds greatly accelerate the rate of oxidation and, accordingly, the rate of formation of all of the various types of oxidation products. The ratio of the amounts of these various types of oxidation products depends on the nature of the oil; therefore, the time required for the formation of a given total amount of oxidation products is measured by the time required for the oil to become corrosive. Some information concerning the predominating type of products of oil change which may be expected in automotive service is obtained. The modified Underwood rates the SAE 20 oils in the order of descending merit as: 4, 1, 2, 5, and 3; and the SAE 30 grade as: 4, 2, 1, 3, and 5. It is interesting to note that this test rates these five oils roughly in the same order as does the oxygen-absorption test.

Engine Stands – Results of tests of these oils on one of the Group F engines are shown graphically in Fig. 3. The SAE 20 grades were used in these tests. The engines operated on stand tests required an SAE 20, and the road test called for an SAE 30; therefore, the grade of oil most suitable to each type of equipment was used. Use of two SAE grades also permits observation of constancy of performance characteristics.

Table 4 shows the results of examination of used oil samples from tests of Oil I on these six engines. This table will show the rate of formation of the various deterioration products during the test of this one oil on each engine. Only the average amounts of each of the deterioration products found in the five samples taken at 1000, 2000, 3000, 4000, and 5000 miles for each of the five oils tested in each of the six engines are shown in Table 5. All the test data on Oil I are given an arbitrary rating at 100%, and then the degree of change in each respect with the other oils is shown as a percentage of the change in Oil I.

Table 6 lists an average of all of the different percentages for each oil on all six of the test engines. Iron content percentages were not included in this summary, since extremely small amounts of iron resulted in a large difference in percentage for the various oils and seemed misleading.

[7] See SAE Transactions, September, 1938, pp. 385-392: "Automotive Bearing Materials and Their Application," by Arthur F. Underwood.

Table 5 – Average of Products of Change Formed at 1000, 2000, 3000, 4000, and 5000 Miles in Oils I, II, III, IV, and V by Tests in Mechanical Laboratory Engines

Group F – Engine No. 1

	Oil I		Oil II		Oil III		Oil IV		Oil V	
		%		%		%		%		%
Neutralization No.	5.1	100	1.1	21	5.2	100	4.8	94	3.7	73
Sludge, %	2.2	100	0.5	23	0.5	23	5.2	236	1.9	86
Chloroform Insoluble, %	1.9	100	0.5	26	0.5	26	0.9	47	1.6	84
Viscosity Increase at 210 F, SSU	46.6	100	1.7	4	108.9	234	61.9	135	28.4	61
Carbon Residue, %	3.7	100	0.9	24	5.8	156	4.5	122	3.9	105
Carbon Residue, Filtered, %	5.9	100	0.4	7	7.6	128	2.4	41	2.8	48
Resin, %	15.03	100	4.00	27	26.49	176	17.39	115	14.97	99
Iron-Drain as Fe, %	0.037	100	0.038	100	0.032	87	0.032	87	0.019	52
Iron-Drain Filtered as Fe, %	0.032	100	0.010	31	0.030	94	0.010	31	0.004	12
True Color	2287	100	322	14	8494	372	2508	110	2760	122

Group F – Engine No. 2

	Oil I		Oil II		Oil III		Oil IV		Oil V	
		%		%		%		%		%
Neutralization No.	4.3	100	3.0	70	4.5	105	4.0	93	4.2	98
Sludge, %	1.7	100	1.1	65	1.7	100	2.8	165	2.1	124
Chloroform Insoluble, %	1.4	100	0.9	64	0.3	21	0.7	50	1.2	86
Viscosity Increase at 210 F, SSU	30.8	100	13.9	45	41.1	137	36.4	118	38.7	125
Carbon Residue, %	2.6	100	1.8	69	4.1	157	3.8	146	4.3	166
Carbon Residue Filtered, %	1.1	100	1.1	100	3.9	354	0.9	82	3.7	336
Resin, %	14.33	100	10.66	74	18.03	126	13.77	96	17.86	125
Iron-Drain as Fe, %	0.019	100	0.014	74	0.018	95	0.028	147	0.025	131
Iron-Drain Filtered as Fe, %	0.007	100	0.008	114	0.017	242	0.010	143	0.006	86
True Color	783	100	886	115	4353	557	2036	260	3626	463

Group G – Engine No. 1

	Oil I		Oil II		Oil III		Oil IV		Oil V	
		%		%		%		%		%
Neutralization No.	5.3	100	2.0	38	5.4	102	3.6	68	3.4	64
Sludge, %	1.2	100	0.4	33	0.4	33	0.6	50	0.8	67
Chloroform Insoluble, %	0.7	100	0.4	57	0.4	57	0.5	71	0.5	72
Viscosity Increase at 210 F, SSU	37.5	100	2.4	6	38.9	102	8.3	22	15.9	43
Carbon Residue, %	2.3	100	1.8	78	4.9	212	2.8	123	3.2	138
Carbon Residue, Drain Filtered, %	1.5	100	2.4	160	6.1	407	0.9	60	2.1	140
Resin, %	18.25	100	6.35	35	22.00	120	6.64	56	10.94	60
Iron-Drain as Fe, %	0.007	100	0.035	500	0.036	515	0.041	585	0.020	286
Iron-Drain Filtered as Fe, %	0.006	100	0.031	518	0.025	417	0.024	400	0.010	167
True Color	677	100	1708	252	5416	800	1624	240	2742	406

Group G – Engine No. 2

	Oil I		Oil II		Oil III		Oil IV		Oil V	
		%		%		%		%		%
Neutralization No.	5.1	100	4.0	79	3.7	72	3.4	67	2.7	53
Sludge, %	0.9	100	0.9	100	0.3	33	0.6	67	0.8	89
Chloroform Insoluble, %	0.6	100	0.7	117	0.3	50	0.5	82	0.3	50
Viscosity Increase at 210 F, SSU	33.8	100	10.3	30	11.2	33	7.2	21	11.0	30
Carbon Residue, %	2.5	100	2.3	92	3.3	132	2.5	100	2.7	108
Carbon Residue, Drain Filtered, %	1.8	100	2.9	162	3.7	206	1.1	61	1.6	89
Resin, %	17.56	100	9.29	53	10.10	57	7.78	44	8.68	50
Iron-Drain as Fe, %	0.008	100	0.051	638	0.013	163	0.067	840	0.007	88
Iron-Drain Filtered as Fe, %	0.006	100	0.036	600	0.011	186	0.051	850	0.004	67
True Color	846	100	1956	232	3386	400	1237	146	2186	259

Group H – Engine No. 1

	Oil I		Oil II		Oil III		Oil IV		Oil V
		%		%		%		%	
Neutralization No.	6.3	100	4.3	88	6.3	100	4.1	65	Engine H-1 did no complete test due to high oil consumption
Sludge, %	2.6	100	1.2	46	0.5	19	2.0	77	
Chloroform Insoluble, %	2.5	100	1.1	44	0.5	20	0.6	24	
Viscosity Increase at 210 F, SSU	48.5	100	20.5	42	148.1	306	23.9	49	
Carbon Residue, %	3.6	100	2.8	78	6.3	175	2.4	67	
Carbon Residue, Drain Filtered, %	2.2	100	2.6	117	9.3	425	0.9	41	
Resin, %	16.53	100	11.69	71	30.48	185	10.34	83	
Iron-Drain as Fe, %	0.053	100	0.055	104	0.057	108	0.030	57	
Iron-Drain Filtered as Fe, %	0.020	100	0.042	210	0.050	250	0.013	65	
True Color	1226	100	1597	130	5339	437	605	49	

Group H – Engine No. 2

	Oil I		Oil II		Oil III		Oil IV		Oil V	
		%		%		%		%		%
Neutralization No.	4.7	100	5.9	125	10.7	220	4.8	104	5.0	106
Sludge, %	0.7	100	2.1	300	0.5	71	2.8	400	3.3	470
Chloroform Insoluble, %	0.5	100	1.8	360	0.5	100	1.0	200	1.7	340
Viscosity Increase at 210 F, SSU	27.3	100	20.9	77	264.6	970	45.8	168	74.1	272
Carbon Residue, %	2.2	100	3.1	140	6.6	300	3.9	177	5.5	250
Carbon Residue, Drain Filtered, %	2.4	100	2.5	104	8.9	370	2.0	83	7.6	312
Resin, %	16.7	100	15.92	99	39.72	242	17.68	108	19.03	118
Iron-Drain as Fe, %	0.011	100	0.027	244	0.030	270	0.031	280	0.042	380
Iron-Drain Filtered, as Fe, %	0.010	100	0.019	190	0.033	330	0.016	160	0.037	370
True Color	612	100	1240	203	5346	870	1533	246	5323	870

Table 6 – Summary of Percentage of Oil Deterioration Products in All Engines Based on Oil I

	Oil I	Oil II	Oil III	Oil IV	Oil V
Neutralization No.	100	70	116	82	79
Sludge, %	100	94	46	166	167
Chloroform Insoluble, %	100	111	46	79	126
Viscosity Increase at 210 F, SSU	100	34	297	86	106
Carbon Residue, %	100	80	189	122	153
Carbon Residue (Drain Filtered), %	100	108	315	61	177
Resin, %	100	60	151	80	90
True Color	100	154	573	208	424
Average	100	89	216	111	167

Table 7 – Summary of Percentage of Oil Deterioration in Each Engine Based on Oil I

	Oil I	Oil II	Oil III	Oil IV	Oil V
Group F, Engine 1	100	19	152	112	85
Group F, Engine 2	100	75	195	126	190
Group G, Engine 1	100	82	228	84	124
Group G, Engine 2	100	108	123	74	91
Group H, Engine 1	100	77	208	72	x
Group H, Engine 2	100	176	393	173	342
Average	100	89	213	107	166

Table 8 – Engine Cleanliness

	Oil I		Oil II		Oil III		Oil IV		Oil V	
Engine	Engine Average	%	Engine Average	%	Engine Average	%	Engine Average	%	Engine Average	%
F-1	21	100	14	67	14	67	28	133	30	143
F-2	27	100	19	71	14	52	26	97	27	100
G-1	55	100	19	35	16	29	21	38	24	44
G-2	57	100	19	33	18	32	20	35	25	44
H-1	27	100	15	55	13	49	23	85	x	x
H-2	49	100	25	51	15	31	37	75	33	67
Average		100		52		43		77		80

From Table 6, we find the tendency of Oils IV and V to form oil-insoluble products with high carbon residue and to become highly colored, and Table 8 also shows that these oils formed rather heavy engine deposits in all three engine groups, especially in engine group F.

Table 6 indicates that Oil III forms oil-soluble products resulting in great increases in oil viscosity, high carbon residue of the sludge-free oil, and marked oil discoloration. Comparison with Table 8 shows Oil III resulted in the cleanest engine interior and the dirtiest oils, suggesting excellent detergent properties at the expense of fluidity and stability. Engines H-1 and H-2 produced with Oil III an average viscosity increase from an original of 55 SSU at 210 F, to 203 and 319 SSU at 210 F, respectively, with final viscosities nearly double the averages; while engine groups F and G produced less severe increases.

The average of the deterioration products from all engines determined by examination of the used oils as shown in Table 6, rates the oils in decreasing order as 2, 1, 4, 5, and 3.

Table 7 shows the average for all the deterioration products for each oil as measured by each engine based on the percentage change below or above Oil I, and these data also rate the oils in descending order of merit as 2, 1, 4, 5, and 3.

Table 8 shows the numerical engine cleanliness rating of each engine operated with each oil and the percentage comparison of the five oils. The engines were rated at the end of each test run by an experienced engineer using retained engine parts as standards. These ratings are made on the scale of 0 to 100 with a clean part being rated as 0 and one sufficiently dirty to make operation impossible being rated as 100. A typical engine inspection sheet is shown in Fig. 4. From these inspection sheets, the engine average for each engine after operating with each of the five oils is shown and compared on a percentage basis.

Engine group G rates Oil I materially worse than do groups F and H, but all engines show that Oil I produced considerable engine deposit. All three engine groups show that Oils II and III are quite free of deposit characteristics and that Oils IV and V produce moderately heavy deposits. The data reveal the importance of using several engine designs in evaluating oils in service, since one type of engine can rate oils in materially different order than the average of three types of engines.

Table 9 shows a detail of the numerical ratings obtained on the more important engine parts with respect to the manner in which individual engines affected each oil. An oil that showed heavy skirt deposits on one type piston

Table 9 – Location of Important Engine Deposits as Taken from Engine Rating Sheets

Engine	Location	Oil I	Oil II	Oil III	Oil IV	Oil V
F-1	Valve gallery	20	15	15	35	25
	Engine interior	25	15	15	35	25
	Oil pan	20	20	25	30	30
	Oil screen	10	0	0	25	5
	Piston skirt	15	5	5	25	45
	Ring grooves	20	20	25	25	70
	Valve stem	20	10	10	10	20
F-2	Valve gallery	25	10	15	35	25
	Engine interior	20	15	15	30	25
	Oil pan	40	25	25	30	25
	Oil screen	40	0	5	10	15
	Piston skirt	25	15	5	25	30
	Ring groove	30	25	30	35	70
	Valve stem	25	10	10	15	15
G-1	Valve gallery	55	25	10	20	20
	Engine interior	65	25	10	20	25
	Oil pan	75	25	20	20	25
	Oil screen	95	0	5	0	5
	Piston skirt	55	35	20	25	30
	Ring groove	85	25	35	35	40
	Valve stem	80	10	5	15	20
Engine	Location	Oil I	Oil II	Oil III	Oil IV	Oil V
G-2	Valve gallery	60	20	20	20	25
	Engine interior	65	20	20	15	25
	Oil pan	75	15	15	20	35
	Oil screen	95	0	0	0	10
	Piston skirt	60	35	20	25	30
	Ring groove	85	35	35	30	35
	Valve stem	85	5	10	10	25
H-1	Valve gallery	45	30	20	30	Did
	Engine interior	30	15	15	25	not
	Oil pan	25	10	15	20	finish
	Oil screen	25	0	0	0	test
	Piston skirt	15	5	5	20	....
	Ring groove	35	10	5	35	....
	Valve stem	50	5	15	15	....
H-2	Valve gallery	80	45	20	55	45
	Engine interior	70	30	20	50	35
	Oil pan	45	20	20	50	25
	Oil screen	95	45	5	30	30
	Piston skirt	20	5	5	20	10
	Ring groove	75	45	20	70	70
	Valve stem	65	5	10	15	30

							Date	7-4-40
							Run No.	44

Engine Make and No.	Group F-No. 1						Engine Deposits	
Date Started	6-26-40						Location	Rating
Date Completed	7-1-40						Valve Gallery	20 SL
Oil Used	Oil No. 1						Crank Shaft	25 CL
Miles Break-In	500						Engine Interior	25 SL
Miles on Test	5000						Oil Pan	20SLC
Speed - M.P.H.	60 avg.						Oil Pump Screen	10 C
Load - H.P.	28 avg.						Piston Skirt	15 L
Water Outlet Temp. °F.	185°						Piston Under Head	10 C
							Piston Head	30C Lead
							Cyl. Below Ring Travel	20 L
	Start	Finish	Avg.				Valve Deck	35C Lead
Oil Gallery Temp. °F.	270	283	279				Valve Stem	20 CL
Oil Pressure	14	15	14.8				Connecting Rod Bearings	35 L
OIL ADDED							Piston Ring Grooves	20 C
Miles	Amount	Miles	Amount	Miles	Amount		Connecting Rods	15 L
3212	1 qt.						Wrist Pins	10 L
							Cylinder Head	30C Lead
							Engine Average	21
							Stuck Rings in Tenths	0

Sample Miles	1000	2000	3000	4000	Drain 5000		
Sludge % by Wt.	0.15	0.20	1.82	3.28	5.54		
N.N. Sludge Present	0.5	4.3	6.0	6.4	8.5		
C.R. Sludge Present	0.41	1.91	4.97	4.90	6.40		
Vis.@ 210°F. Sludge Present	54.7	66.4	106.2	116.2	161.0		
Resin Per Cent	2.18	13.80	24.08	18.31	16.82		

See reverse side for general remarks.
Engine deposits C,L,S, stand for carbon, lacquer, and sludge respectively. Under ratings the lower numerical value indicates the cleanest engine, ratings are made from 0 to 100.

■ Fig. 4 – Typical engine inspection sheet

showed moderate to light deposits on a different type piston. Oils that one valve design rates as objectionable, another valve design will rate acceptable.

From the standpoint of engine cleanliness alone (Table 8), the engines rate the oils from best to poorest as 3, 2, 4, 5, and 1.

Comparison of data from Tables 6, 7, and 8 shows Oil III producing the cleanest engines and the dirtiest used oils, indicating good detergency at the expense of oil stability. Oil I produced the dirtiest engines but rated second in oil cleanliness condition, indicating relatively good oil stability but relatively poor detergency. The other three oils are indicated to possess better balanced performance properties with Oil II showing the best stability and second-best engine cleanliness, Oils IV and V following in order.

Giving due consideration to the equally important factors of oil stability and engine interior cleanliness, we find from Tables 6 and 8:

	Oil I	Oil II	Oil III	Oil IV	Oil V
Table 6 – Oil Condition	100	89	216	111	167
Table 8 – Engine Condition	100	52	43	77	80
Average	100	71	130	94	124

Thus the mechanical laboratory engines show that the rating of the five oils from best to poorest is: 2, 4, 1, 5, and 3.

III. Road Tests

The results of examination of the used oil from each of the four drains from all road tests using Oil II are listed in Table 10. In order to permit comparison of the various

Table 10 – Products of Change in Oil II in Each Oil Drain from Road Tests

	Group A – Engine No. 3					Group B – Engine No. 3				
Drain	1	2	3	4	Ave.	1	2	3	4	Ave.
Miles	2592	2304	2304	2880	2520	2592	2304	2304	2880	2520
Neutralization No.	1.05	0.22	0.15	0.20	0.41	8.5	1.4	1.2	1.4	3.1
Sludge, %	1.11	0.16	0.12	0.09	0.37	1.48	1.31	1.58	1.85	1.56
Chloroform Insoluble, %	0.93	0.07	0.08	0.05	0.28	1.01	0.56	1.29	1.69	1.14
New Oil Viscosity at 210 F, SSU	60.4	60.4	60.4	60.4	60.4	60.4	60.4	60.4	60.4	60.4
Drain Viscosity at 210 F, SSU	59.8	60.0	59.4	60.2	59.9	89.0	61.0	61.0	60.6	67.9
Carbon Residue, %	1.08	0.085	0.086	0.13	0.345	3.95	1.88	1.71	2.34	2.49
Carbon Residue Filtered, %	0.58	0.074	0.042	0.06	0.189	3.24	1.03	0.28	1.25	1.45
Resin, %	1.19	1.22	1.11	0.81	1.08	17.28	3.20	1.68	1.22	5.85
Iron as Fe, %	0.036	0.002	0.001	0.002	0.010	0.030	0.009	0.009	0.008	0.014
Iron as Fe Filtered, %	0.013	0.002	0.001	0.001	0.004	0.024	0.006	0.002	0.006	0.010
Lead as Pb, %	0.129	0.021	None	0.04	0.17	0.649	0.920	0.706	1.28	0.89
True Color	179	42	41	46	77	1460	256	91	120	482
Dilution, %	1.2	1.6	0.8	1.6	1.3	2.4	2.0	1.0	1.6	1.8
Water, %	None	None	Trace	None	None	None	None	Trace	None	None
Flash, F	430	380	390	350	388	400	405	380	370	389

	Group C – Engine No. 3					Group D – Engine No. 2				
Drain	1	2	3	4	Ave.	1	2	3	4	Ave.
Miles	2592	2304	2304	2880	2520	2706	2460	2460	2460	2522
Neutralization No.	5.0	0.5	0.4	0.4	1.6	0.2	0.2	0.4	0.25	0.26
Sludge, %	0.20	0.27	0.14	0.08	0.17	0.15	0.09	0.12	0.05	0.10
Chloroform Insoluble, %	0.11	0.11	0.06	0.01	0.07	0.07	0.05	0.04	0.02	0.05
New Oil Viscosity at 210 F, SSU	60.4	60.4	60.4	60.4	60.4	60.4	60.4	60.4	60.4	60.4
Drain Viscosity at 210 F, SSU	70.2	60.6	59.8	61.0	62.9	55.6	57.0	55.8	56.4	56.2
Carbon Residue, %	2.31	0.38	0.16	0.18	0.76	0.086	0.077	0.07	0.07	0.076
Carbon Residue Filtered, %	2.20	0.21	0.14	0.12	0.67	0.067	0.066	0.062	0.05	0.061
Resin, %	10.30	2.29	1.72	1.33	3.91	0.88	0.89	0.87	1.02	0.92
Iron as Fe, %	0.010	0.004	0.003	0.001	0.005	0.001	0.002	0.002	0.004	0.002
Iron as Fe Filtered, %	0.012	0.002	0.003	0.001	0.005	0.001	0.002	0.002	0.004	0.002
Lead as Pb, %	0.617	0.106	0.07	0.07	0.23	0.006	0.005	0.01	0.02	0.01
True Color	346	76	69	91	146	30	32	24	32	30
Dilution, %	1.8	3.2	1.8	1.2	2.0	3.0	2.0	3.8	2.8	2.9
Water, %	None	None	None	None	None	None	None	Trace	None	None
Flash, F	350	330	340	335	339	270	230	260	255	254

Table 11 – Average Amounts and Comparative Percentages of Products of Change in Oils in Road Tests – 2520 Miles

Group A	Oil I	%	Oil II	%	Oil V	%
Neutralization No.	0.31	100	0.41	132	2.3	740
Sludge, %	0.27	100	0.37	137	0.43	158
Chloroform Insoluble, %	0.19	100	0.28	147	0.27	142
New Oil Viscosity at 210 F, SSU	60.7	....	60.4	....	60.5	....
Drained Oil Viscosity at 210 F, SSU	60.3	100	59.9	91	70.3	254
Carbon Residue, %	0.201	100	0.095	47	2.15	1070
Carbon Residue, Filtered, %	0.057	100	0.189	332	1.85	3240
Resin, %	1.46	100	1.08	74	7.29	500
Iron as Fe, %	0.004	....	0.010	....	0.053	....
Iron as Fe, Filtered, %	0.002	....	0.004	....	0.027	....
Lead as Pb, %	0.20	....	0.17	....	0.51	....
True Color	42	100	77	187	752	1780
Dilution, %	1.8	....	1.3	....	1.8	....
Water, %	00	....	00	....	00	....
Flash, F	410	....	388	....	408	....
Average	....	100	....	143	....	985

Group B	Oil I	%	Oil II	%	Oil III	%
Neutralization No.	8.7	100	3.1	36	7.3	94
Sludge, %	2.13	100	1.56	78	0.79	37
Chloroform Insoluble, %	0.53	100	1.14	208	0.46	87
New Oil Viscosity at 210 F, SSU	60.7	....	60.4	....	67.9	....
Drained Oil Viscosity at 210 F, SSU	85.6	100	67.9	44	110.6	156
Carbon Residue, %	3.25	100	2.47	76	4.44	136
Carbon Residue, Filtered, %	2.05	100	1.45	71	4.02	196
Resin, %	14.25	100	5.85	41	18.40	130
Iron as Fe, %	0.031	....	0.014	....	0.017	....
Iron as Fe Filtered, %	0.012	....	0.010	....	0.013	....
Lead as Pb, %	0.92	....	0.89	....	0.93	....
True Color	429	100	482	112	1965	457
Dilution, %	1.9	....	1.8	....	2.5	....
Water, %	00	....	00	....	00	....
Flash F	366	....	389	....	380	....
Average	....	100	....	83	....	162

Group C	Oil I	%	Oil II	%	Oil IV	%
Neutralization No.	1.0	100	1.6	160	1.7	170
Sludge, %	0.12	100	0.17	142	0.14	117
Chloroform Insoluble, %	0.06	100	0.07	117	0.09	150
New Oil Viscosity at 210 F, SSU	60.7	....	60.4	....	62.0	....
Drained Oil Viscosity at 210 F, SSU	61.1	100	62.9	121	67.5	169
Carbon Residue, %	0.49	100	0.76	155	1.32	270
Carbon Residue Filtered, %	0.42	100	0.67	159	1.13	270
Resin, %	3.39	100	3.91	115	5.72	169
Iron as Fe, %	0.005	....	0.005	....	0.006	....
Iron as Fe Filtered, %	0.003	....	0.005	....	0.005	....
Lead as Pb, %	0.13	....	0.23	....	0.33	....
True Color	103	100	146	142	225	209
Dilution, %	1.3	....	2.0	....	1.6	....
Water, %	00	....	00	....	00	....
Flash, F	318	....	339	....	353	....
Average	....	100	....	139	....	191

Group D 2522 Miles	Oil I	%	Oil II	%	Oil III	%	Oil IV	%
Neutralization No.	0.3	100	0.26	87	3.4	1113	3.12	1080
Sludge, %	0.11	100	0.10	91	0.22	200	0.18	164
Chloroform Insoluble, %	0.07	100	0.05	71	0.12	172	0.09	128
New Oil Viscosity at 210 F, SSU	60.7	....	60.4	....	67.9	....	62.0	....
Drained Oil Viscosity at 210 F, SSU	56.2	100	56.2	92	70.6	388	63.2	326
Carbon Residue, %	0.081	100	0.061	94	2.42	2980	1.83	2260
Carbon Residue Filtered, %	0.057	100	0.061	107	2.30	4040	1.42	2500
Resin, %	1.27	100	0.92	72	7.45	590	6.63	523
Iron as Fe, %	0.002	....	0.002	....	0.006	....	0.008	....
Iron as Fe Filtered, %	0.002	....	0.002	....	0.005	....	0.004	....
Lead as Pb, %	0.11	....	0.01	....	0.65	....	0.45	....
True Color	32	100	30	94	441	1375	169	530
Dilution, %	2.8	....	2.9	....	2.8	....	2.9	....
Water, %	00	....	00	....	00	....	00	....
Flash, F	254	....	254	....	248	....	271	....
Average	....	100	....	77	....	1357	....	939

oils with the elimination of engine differences, the average amount of deterioration products and comparison on a percentage basis in each group of test equipment are shown in Table 11. The lubricating oil dilution in all of these engine tests was found to be approximately 2%. It has been determined that 2% dilution reduces the fresh oil Saybolt universal viscosity at 210 F about 7 sec. Accordingly, 7 sec were deducted from the viscosity at 210 F of the unused oils, and the resulting viscosity used in calculating viscosity increase.

Table 12 shows the engine ratings of each engine group, fuel, and oil consumption rates with the various oils. The compression ring wear figures are shown, as they are of sufficient magnitude to indicate a trend with reasonable reliability. Wear of other parts was so small that the first significant figures range from the fourth to the sixth decimal place, and their inclusion might be misleading. Inclusion or exclusion of the wear data, however, does not alter the results because they are not in conflict with the other results. Engine ratings shown were obtained by the same procedure as previously described for the mechanical laboratory engines.

Table 12 – Engine Cleanliness, Corrosion, Wear, and Fuel and Oil Consumption in Road Tests

Group A	Oil I	%	Oil II	%	Oil V	%
Engine Rating	16.9	100	13.2	78	15.8	93
Bearing Corrosion	No		No		Trace	
Compression Ring-Gap Increase	0.0066	100	0.0072	109	0.0537	814
Oil Used, qt per 1000 miles	1.1	100	1.8	167	3.3	300
Gasoline Used, gal per 1000 miles	57	100	56	98	56	98
Average	....	100	....	113	....	426

Group B	Oil I	%	Oil II	%	Oil III	%
Engine Rating	23.7	100	14.2	60	19.1	81
Bearing Corrosion	No		No		No	
Compression Ring-Gap Increase	0.0086	100	0.0052	60	0.0054	63
Oil Used, qt per 1000 miles	1.0	100	0.85	85	1.4	40
Gasoline Used, gal per 1000 miles	62	100	59	95	63	102
Average	....	100	....	75	....	96

Group C	Oil I	%	Oil II	%	Oil IV	%
Engine Rating	18.9	100	14.9	79	16.7	88
Bearing Corrosion	Light		Trace		Trace	
Compression Ring-Gap Increase	0.0129	100	0.0053	41	0.0054	42
Oil Used, qt per 1000 miles	2.6	100	2.0	78	2.1	83
Gasoline Used, gal per 1000 miles	52	100	53	101	54	102
Average	....	100	....	75	....	79

Group D	Oil I	%	Oil II	%	Oil III	%	Oil IV	%
Engine Rating	10.7	100	7.3	68	15.1	141	17.1	160
Bearing Corrosion	No		No		Yes		Yes	
Zero-Lash Valve Lifters Stuck	3	....	1	....	9	....	11	....
Compression Ring-Gap Increase	0.0048	100	0.0048	100	0.0056	117	0.0054	113
Oil Used, qt per 1000 miles	0.49	100	0.41	84	0.44	90	0.47	96
Gasoline Used, gal per 1000 miles	103	100	101	98	106	103	104	101
Average	....	100	....	87	....	113	....	118

Table 13 – Summary of Percentage Comparison of Oil Change

Increase in Neutralization Number

Test Equipment	Grade SAE No.	Oil I	Oil II	Oil III	Oil IV	Oil V
Modified Underwood	20	100	117	141	77	128
Modified Underwood	30	100	71	150	67	240
Engine F-1	20	100	21	100	94	73
Engine F-2	20	100	70	105	93	98
Engine G-1	20	100	38	102	68	64
Engine G-2	20	100	79	72	67	53
Engine H-1	20	100	88	100	65	....
Engine H-2	20	100	125	220	104	106
Road Group A	30	100	132	....	....	740
Road Group B	30	100	36	94	....	....
Road Group C	30	100	160	....	170	....
Road Group D	30	100	87	1113	1080	....

Conradson Carbon Residue

Test Equipment	Grade SAE No.	Oil I	Oil II	Oil III	Oil IV	Oil V
Modified Underwood	20	100	106	166	79	136
Modified Underwood	30	100	100	130	69	322
Engine F-1	20	100	24	156	122	105
Engine F-2	20	100	89	157	146	166
Engine G-1	20	100	78	212	123	138
Engine G-2	20	100	92	132	100	108
Engine H-1	20	100	78	175	67	....
Engine H-2	20	100	140	300	177	250
Road Group A	30	100	47	....	....	107
Road Group B	30	100	76	136	....	....
Road Group C	30	100	155	....	270	....
Road Group D	30	100	94	2980	2260	....

Formation of Sludge

Test Equipment	Grade SAE No.	Oil I	Oil II	Oil III	Oil IV	Oil V
Modified Underwood	20	100	134	68	81	134
Modified Underwood	30	100	91	100	76	76
Engine F-1	20	100	23	23	236	86
Engine F-2	20	100	65	100	165	124
Engine G-1	20	100	33	33	50	67
Engine G-2	20	100	100	33	67	89
Engine H-1	20	100	46	19	77	....
Engine H-2	20	100	300	71	400	470
Road Group A	30	100	137	....	....	158
Road Group B	30	100	78	37	....	....
Road Group C	30	100	142	....	117	....
Road Group D	30	100	91	200	164	....

Formation of Resin

Test Equipment	Grade SAE No.	Oil I	Oil II	Oil III	Oil IV	Oil V
Modified Underwood	20	100	109	151	67	109
Modified Underwood	30	100	90	180	50	450
Engine F-1	20	100	27	176	115	99
Engine F-2	20	100	74	126	96	125
Engine G-1	20	100	35	120	56	60
Engine G-2	20	100	53	57	44	50
Engine H-1	20	100	71	185	83	....
Engine H-2	20	100	99	242	108	118
Road Group A	30	100	74	....	....	500
Road Group B	30	100	41	130	....	....
Road Group C	30	100	115	....	169	....
Road Group D	30	100	72	590	523	....

Increase in Viscosity at 210 F, SSU

Test Equipment	Grade SAE No.	Oil I	Oil II	Oil III	Oil IV	Oil V
Modified Underwood	20	100	108	176	77	141
Modified Underwood	30	100	88	200	67	300
Engine F-1	20	100	4	234	135	61
Engine F-2	20	100	45	137	118	125
Engine G-1	20	100	6	102	22	43
Engine G-2	20	100	30	33	21	30
Engine H-1	20	100	42	306	49	....
Engine H-2	20	100	77	970	168	272
Road Group A	30	100	91	....	....	254
Road Group B	30	100	44	156	....	....
Road Group C	30	100	121	....	169	....
Road Group D	30	100	92	388	326	....

Formation of Color

Test Equipment	Grade SAE No.	Oil I	Oil II	Oil III	Oil IV	Oil V
Engine F-1	20	100	14	372	110	122
Engine F-2	20	100	115	557	260	463
Engine G-1	20	100	252	800	240	406
Engine G-2	20	100	232	400	146	259
Engine H-1	20	100	130	437	49	....
Engine H-2	20	100	203	870	246	870
Road Group A	30	100	187	....	....	1780
Road Group B	30	100	112	457	....	....
Road Group C	30	100	142	....	209	....
Road Group D	30	100	94	1375	530	....

Because of the unavoidable differences between mechanical laboratory engine tests and road tests, the possibility of making an equitable or acceptable numerical comparison of the results is questioned. The laboratory engines were run 5000 miles without a drain, samples for progressive oil condition change were secured at 1000-mile intervals, providing data for curves, and all engines were thoroughly cleaned between runs. The road engines were drained approximately every 2500 miles, and some oil deterioration products were unavoidably left in the engines to contaminate, to some degree, the succeeding oil charge; and, since samples were taken only at drain periods, final oil condition characteristics only are available. All mechanical laboratory engine tests were made without the use of oil filters. Standard factory-equipment oil filters of three different makes were used on engines of road groups A, C, and D, while group B units were not equipped with filters. Laboratory engines were required to complete 5000 miles of continuous operation while road units were operated during the day only.

Table 14 – Summary of Engine Rating Sheet

Test Equipment	Grade SAE No.	Oil I	Oil II	Oil III	Oil IV	Oil V
Engine F-1	20	100	87	87	136	146
Engine F-2	20	100	71	52	97	100
Engine G-1	20	100	35	29	38	44
Engine G-2	20	100	33	32	35	44
Engine H-1	20	100	55	49	85	....
Engine H-2	20	100	51	31	75	67
Road Group A	30	100	78	....	....	93
Road Group B	30	100	60	81	....	....
Road Group C	30	100	79	....	88	....
Road Group D	30	100	68	141	160	....

Table 13, however, gives a summary of the more important oil changes for all tests, and Table 14 gives a summary of all engine ratings.

An examination of the data in Tables 10, 11, 12, and 13, however, shows the same general oil behavior characteristics; and Table 14 discloses the same engine condition characteristics of the five oils as were indicated from the oxygen-absorption, modified Underwood, and the mechanical laboratory engine tests previously discussed, viz.:

Oil I shows the same relatively good oil stability characteristics as evidenced by low neutralization number, low sludge, low chloroform insolubles, low viscosity increase, low discoloration, and low resin and its lack of detergent properties by high engine condition ratings or engine dirt.

Oil III shows the same relatively high neutralization number, very low sludge and chloroform insolubles, but high viscosity increase with high discoloration, high carbon, and high resin content indicating good peptizing or detergency characteristics at the expense of stability as previously indicated on Tables 6, 7, and 8. These characteristics are more pronounced in engine group B, where oil temperatures were higher than in engine group D. The engine inspections showed relatively good freedom from sludge but considerable resinous deposits, particularly in the hydraulic valve lifters of engine group D where 9 of the 12 lifters were stuck with lacquer.

Oil IV displayed more balanced stability and engine dirt characteristics than Oils I and III, having moderate neutralization number increase, low sludge, low chloroform insolubles, low discoloration, and moderately low carbon residue but high resin content. This oil condition was evidenced by having 11 out of 12 hydraulic valve lifters stuck with resinous deposits.

Oil V is indicated as developing high neutralization number, moderately low sludge, and chloroform insolubles, but having tendencies to increase considerably in viscosity and darken in color with high carbon residue and moderately high resin. Oil V formed sufficient hard carbon on the land above the upper compression rings to act as an abrasive. This probably was the cause of the exceedingly high increase in piston-ring gaps which, in turn, promoted high oil consumption. The car using Oil V detonated badly during acceleration, indicating increased compression ratio due to carbon deposits.

Oil II again shows to advantage with low neutralization number, low sludge, low chloroform insolubles, low viscosity increase, low discoloration, and low carbon residue and low resin, and the engine inspections showed the best ratings of all oils tested.

The effect of increased viscosity of Oils III, IV, and V as a result of oil soluble contaminating materials is interesting, as evidenced by poorer fuel economy and generally increased oil consumption indicating less effective ring action. Lacquer deposition is noted with Oils III and IV, especially in the hydraulic valve lifters, indicating that these oils had approached or just passed the critical point and that resins were dropped out of solution. This result suggests the probability that the oil soluble metallic catalysts left in the engines at the drain periods were sufficient to accelerate oil decomposition and the resultant deposition. This probably accounts for greater oil punishment by road test units than by the engine stands as observed by Sibley et al[1], in spite of lower operating temperatures of road-test units.

Each of the more significant oil characteristics as shown in Table 11 has been evaluated, listing the oils in order of descending merit with respect to each characteristic. This study rates the oils as follows:

Neutralization Number	2	1	3-4		5
Sludge, %	1	2	3	4	5
Viscosity (drained oil), SSU	2	1	4	3	5
Carbon Residue, %	2	1	5	4	3
Resin, %	2	1	3-4		5
True Color	1	2	4	3	5
Oil, Order Descending Merit	2	1	4	3	5

In the same manner, from Table 12, the major factors of engine condition during and after test have been rated individually and show:

Engine	2	3	5	1	4
Bearing Corrosion	2-1		3-4		5
Compression Ring Gap Wear	2	1	4	3	5
Oil Consumption	2	1	3	4	5
Gasoline Consumption	2	5	1	4	3
Engine, Order Descending Merit	2	1	3	4	5

Giving equal consideration to oil stability and engine condition, we find the order of descending merit to be:

Oil Stability	2	1	4	3	5
Engine Condition	2	1	3	4	5
Overall Order Descending Merit	2	1	3-4 4-3		5

■ Conclusion

Thus, the oxygen-absorption test, modified Underwood test, mechanical laboratory engine, and road-test units rate the five oils in order of descending merit as:

Oxygen Absorption Test	1	2	4	3	5
Modified Underwood Test SAE 20	4	1	2	5	3
Modified Underwood Test SAE 30	4	2	1	3	5
Mechanical Laboratory Engines SAE 20	2	4	1	5	3
Road-Test Engines SAE 30	2	1	3-4 4-3		5

From the study of these five oils, considering both the nature and degree of oil change and engine conditions, the data indicate that Oil II should give outstandingly good results in commercial use. The accuracy of this indication has been adequately proved by substantially a year's commercialization of the lubricant.

The mass of data accumulated and an analysis of the overall picture of results certainly indicate that there is still considerable room for debate as to the optimum degree of peptizing or detergency desired in an oil. The minimum change in oil characteristics in service is most imperative along with enough peptizing to prevent excessive deposition of materials of either intrinsic or extrinsic origin. The evidence certainly indicates the desirability of preventing the formation of products of oil change rather than permitting their formation and causing them to be dissolved in the oil, since such oil-soluble products result in accelerated rates of oil change due to their catalytic effect.

Since it has been shown that the road-test units displayed higher rates of oil change even though they had lower operating temperatures, it appears that the catalytic effect of oil deterioration residues left in the engines at oil-change periods is of greater magnitude than is generally recognized; and it is doubtful if increased engine-stand oil temperature can be substituted satisfactorily for this effect. It is recognized that engine-stand operations should always start with a scrupulously clean engine to avoid the "poisoning" effect of residues on the next oil tested. Otherwise, the observed results are almost invariably misleading. The work indicates the desirability of consideration of this factor in engine-stand tests.

This series of tests indicates the indispensability of a variety of equipment in oil testing and the futility of hope that a single piece of equipment may be developed which will quickly and correctly evaluate lubricating oils. This is further shown by the fact that even this comprehensive series of tests failed to provide the opportunity to confirm or refute the indicated corrosive tendency of the SAE 30 grade of Oil V shown by the modified Underwood test in Table 3.

The Automotive Body Engineer in AIRCRAFT

by JOHN C. WIDMAN

Aircraft Division, The Murray Corp. of America

WHEN the automotive engineer enters the aircraft industry, he is confronted immediately with different engineering methods, with new and strange materials, and with unfamiliar manufacturing methods and process operations which are in use with these materials, and he must become thoroughly conversant with these differences and the reasons for them before he can undertake to make a valid contribution, is the contention of Mr. Widman.

"I know of no phase of engineering in which a varied knowledge is so essential as it is in aircraft," he says, and he lists as desirable some knowledge of heat-treatment, structural and electrical engineering, hydraulics, metallurgy, and chemistry.

Mr. Widman contrasts the automotive and aircraft industries throughout his paper and concludes that the present war experience will have a lasting effect upon the automotive industry, perhaps leading to the entrance of the automobile manufacturer into the aircraft field.

THE AUTHOR: JOHN C. WIDMAN was made product engineer upon the formation of the Aircraft Division, The Murray Corp. of America, in 1940. Mr. Widman started to work for The Murray Corp. of America by designing and building a water glider to be towed behind a speed boat. The project was successful, but was dropped due to the depression. Mr. Widman held many production assembly positions with The Murray Corp. of America prior to his appointment as product engineer. He graduated from the University of Michigan in 1930, in aeronautical engineering.

■ ■ ■

IN a comparison of the fundamental difference between aircraft engineering and automobile engineering, one should go back into the origin of the two branches and note some of the basic characteristics of each.

The methods and terminology of the automotive phase is a clue to its origin. In auto work we are accustomed to speak of such things as running boards and dash panels. We spoke of mud guards long before we called them fenders. Some might even remember when cars had whip sockets. As we all know, the original engineers in the automotive field were carriage makers.

■ Shipbuilding Terms Used

Aircraft-engineering methods likewise can be traced by the terms in common use today. The aircraft industry drew some of its original personnel from the shipbuilding trade and consequently we hear of bulkheads, water lines, buttock lines, and loft boards.

Although both the automotive and aircraft industries had their start in the same general part of this country at approximately the beginning of the century, the automobile had the advantage of a more rapid advancement. It more quickly took hold of public fancy and consequently boomed into one of the leading industries of the world. Thus it was natural that the methods employed in the automotive-engineering field developed around the idea of mass production.

Many of us have worked in an automotive engineering department which was such in name only. Upon approval of a design, a few lines were laid down on a sheet of paper to control the shape and locate the windows and doors. This draft was transferred to the shop. The shop layout man took this sketch and from it laid out all his own pillars and frame work and framed his own body. It was then paneled, painted, and trimmed, all by an expert in his own field who had neither a template nor a drawing to aid him or distract him from his job. It was only in later years, when mass production demanded the specialist who spends full time on one particular job, that it became necessary to place in blueprints and specifications the information required to enable someone else to produce similar parts in distant localities.

Automotive companies pride themselves that their blue-

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prints can be sent all over the world and the parts described therein readily duplicated, and many of them do successfully follow this practice.

But now, let us take a glimpse at the development in the aircraft industry. Until approximately ten years ago, it was impossible for any aircraft plant to count on building more than a comparatively few planes of any one design. Only a few companies were able to obtain sizable government contracts. Only a few were able to supply the expanding airlines trade with the type of plane that the public demanded. The majority of manufacturers had to be content with "10 of this" and "25 of that." It was only natural that engineering costs under such conditions should be held to the absolute minimum.

It followed that a condition was built up where the drawings were, in some cases, little more than pictures or diagrams. From this condition it was natural that the information on the drawing must be supplemented in the form of templates, tools, and so on. Matters were further complicated because it has seldom been practical or economical to go back and incorporate shop changes in the blueprints.

Information from Many Sources

And so the automotive engineer, upon his introduction to aircraft, is confronted with information provided him on drawings, on templates, and on tools. All of this information is necessary; yet none of it is complete in itself, and some of it is conflicting.

This condition is the same as that which existed in some phases of the automobile industry a few years ago. Not having the advantage of amortization of engineering costs over a large number of units, the industry naturally adopted a system whereby only the most meager information was issued to experienced mechanics who were to build the product. Such a procedure recalls the custom body shop in the past.

The aircraft industry does not favor a system of one engineering department, but generally has two departments which furnish engineering information.

First there is the department known as the "Engineering Department." Here they lay out the general design of the ship – making assembly drawings which are usually to ¼ or ½ scale and contain overall dimensions, locating dimensions, rivet call-outs, bolt sizes, and general information of that nature. The Engineering Department also draws castings, forgings, machined parts, and such accessories as the landing gear and its operating mechanism and door-closing mechanisms. This department also works out the many and varied operating and control details. These functions generally cover the scope of the Engineering Department.

Obviously these are very necessary and important functions but also we know that it is not the complete engineering picture. Information as to the detail layout of individual parts such as stampings, extrusions and rolled sections must be secured from the "Loft Department."

The Loft is a separate department. While sometimes a part of the tool division and sometimes a part of the engineering division, it is always distinct and separate from the engineering department.

In the Loft the engineering blueprints showing the general structure design are laid down full size on plates which are known as "Loft Boards" or "lines boards." The Loft develops its own surfaces from the basic design and cuts its own sections and develops the detail parts as indicated by the engineering blueprints. Different sections or "stations," as they are generally called, usually have a separate loft board. For example, each of the bulkheads of a nacelle or a fuselage is on a separate board, as is each of the ribs assemblies of a wing. This procedure differs sharply from the automotive practice of one master layout or "draft."

The individual station boards are then used for the layout and making of templates and, of course, it is the template which is the basic unit of detail information in the aircraft industry.

From a combination of these factors the shop must gather its information for the manufacture of tools and parts.

Another general practice is to use a standards book cataloguing innumerable small parts, and to call out on the drawing for a certain part number which is generally known to be listed in this standards book.

In both industries the layout problems are approximately the same. There are a few more parts in an airplane, of course, and this is purely a matter of additional personnel. Automotive men make detail drawings which contain all the information necessary. Aircraft practice is to have information partially on the drawings and partially on templates.

I do not mean to criticize these practices. It is merely my intention to point out that they are different and strange to men who have worked with methods used in automotive plants and that these men require education and closer supervision than if they were working in more familiar roles.

When one realizes that we are dealing with an industry which has expanded enormously in the last few years despite a great scarcity of engineering talent, it is not surprising that the engineering department has had to confine its efforts to basic designs leaving detail information to other departments.

New Materials Bring Problems

In addition to different engineering methods, the automotive man is confronted with new and strange materials and unfamiliar manufacturing methods and process operations which are in use with these materials.

By far the most common material used in aircraft today is aluminum.

Aluminum differs considerably from steel in its handling and methods of manufacture. The first thing one learns is that "scratched aluminum is scrapped aluminum." Scratches of course are undesirable in any finished product. Aluminum being softer than steel, it scratches more readily. Scratches are a structural detriment inasmuch as in thin sheets they cause a reduction in section and are also a focal point for starting of cracks. These cracks can cause failures in the structure due to the relatively high stresses under certain conditions of flight.

Scratches in Alclad material, even though not very deep, are likely to cut through the protective coating of pure aluminum into the alloy and cause corrosion. Obviously these cracks are of such a serious nature that handling and manufacturing methods must be such as to produce parts as "scratch-free" as possible.

The use of rubber dies and "Guerin" blocks is not only a relatively cheap method of manufacture but also beneficial in that scratches are eliminated. One, however, learns that the rubber die has its limitations and the design of parts must be modified accordingly.

The limit to which aluminum can be worked is much narrower than that to which steel may be subjected. It does not draw or stretch as readily as steel. There are very definite limits to which it can be bent or formed. It is also a work-hardening material. It differs considerably from steel in its heat-treatment.

In all phases of engineering it is desirable to have considerable knowledge of the phases of engineering other than the particular branch in which we are working. This is particularly so of the aircraft engineer. I know of no phase of engineering in which a varied knowledge is so essential as it is in aircraft.

The heat-treatment of materials is a subject on which the aircraft man must be well versed both in ferrous and non-ferrous alloys.

Structural engineering plays a prominent part in the determination of loads, the analyzing of stresses, and specification of structures to support these loads properly, the proper placement of rivets, and the proper bending and forming of parts so as not to exceed their elastic limit.

In modern aircraft such mechanisms as bomb doors, landing-wheel doors, the landing gear, brakes, landing flaps, and similar items are hydraulically operated and, therefore, some knowledge of hydraulics is useful.

There are innumerable electrical installations in an airplane and it is helpful to have some knowledge of this subject, even though there are electrical engineers available.

In addition, a knowledge of metallurgy is very important in these days of priorities and substitutions. The proper use of materials, substitution of materials, hardness scales, and similar problems are part of the every-day life of the aeronautical engineer.

For the chemical engineering phase we deal in problems of cleaning metals, corrosion-resisting treatments, paints, primers, spark-inhibiting finishes, etching, and similar subjects, some of which are familiar to the automotive engineer.

■ Government Specifications

Government specifications and regulations are other matters which are new to the automotive man. The ultimate purchaser of the aircraft which are being produced today in the automotive plants is the military service of the United States Government. The government and its many agencies have numerous publications specifying certain conditions which must be met. These regulations, however, are really helpful since, in them, a newcomer to the field is able to find much information on a range of subjects. All of this is extremely valuable to him in his effort to learn as much as he can in a short time. It is rarely possible to find so much extensive information in such compact form, outside of government work.

I believe it is the place of the automotive man to learn in advance as much as possible of the afore-mentioned problems, to study the production problems and the methods used in the aircraft industry today and, only after this careful study, to apply the productive methods with which he is familiar.

This procedure will enable him better to understand the results for which the aircraft man is striving. Until he understands the aims and methods of the aircraft man, he cannot expect to produce the desired result equally well. By familiarizing himself with the methods of the aircraft man, the automotive man can, with better advantage, transfer his knowledge of high-productive methods and high-productive equipment to the aircraft industry.

It is in this latter phase that the talents of the automotive industry can be placed to utmost advantage in this critical period. We need airplanes and airplane parts in quantities. We of the automotive industry can produce them if we realize that there are others who have had considerable experience in producing these articles and that we can benefit by their experience and knowledge.

Based on this point of view we can best apply the talents for which Detroit is famous.

■ Aluminum Knowledge Required

The automotive man should learn as much as possible of aluminum, its advantages and its disadvantages. He should review or study the various subjects just mentioned. No one will become an expert in all these fields, but a working knowledge and an ability to put one's hands on pertinent information when needed is a valuable asset. Should one wish to specialize in certain branches, such as hydraulics, structures or electrical, he certainly can find his place in the aircraft industry. The automobile body full-size layout man, after a few months of becoming familiar with aircraft standards and regulations, can turn out a first-rate job.

It is doubtful if the average automobile plant can be converted so easily. Units of the size of modern aircraft or their major assemblies require such space for fabrication that many of our buildings are seriously cramped. In some cases it is necessary to construct new buildings, especially for final assemblies. Sub-assemblies can, however, be manufactured in existing buildings with only slight alterations.

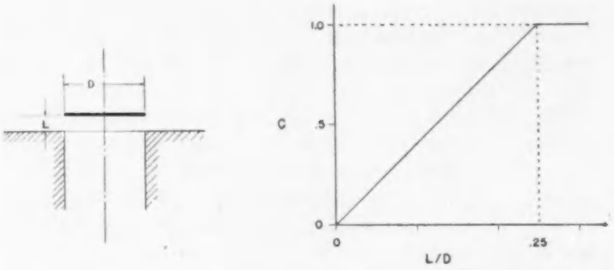
Equipment in automobile plants can be utilized to a great extent, especially equipment for heavy formed parts. Lighter parts are generally made on hydraulic presses with the rubber pad. Routing, high-frequency spot-welding, chemical tanks for anti-corrosion processes, and a large amount of drilling and riveting are operations for which the average auto plant is not equipped.

Some substitutions can be made. For instance, blanking dies can generally be substituted for the routing operation if production warrants their manufacture. Certainly all machine-tool equipment is extremely valuable.

It is not beyond reason to suppose that this concentrated course in aircraft, which the automobile industry is now undergoing, will have a lasting effect in the industry. I believe that some of the larger automotive manufacturers have been contemplating for some time the entrance into the aircraft field. By the time this national emergency is over, there will be a large number of trained personnel available for future use. With this personnel available, some of the manufacturers will undoubtedly wish to remain in the industry.

The automotive companies have always had a highly competitive market. There is no doubt that, with new plants, equipment, and personnel available, they can attempt to take their place with others.

A I R   F L O W



■ **Fig. 1 – Basis for intake-valve coefficients**

IT has long been recognized by designers of internal-combustion engines that volumetric efficiency, and therefore power output, may be increased by improvements in the flow characteristics of the intake system. Many studies of different valve and port combinations have been made, but this work is usually done as a means of improving the performance of a specific engine design by the method of cut and try. An estimate of the possible improvement would be useful to determine the need for, or value of, such research.

Analysis of the charging process of the four-stroke cylinder is complicated by the simultaneous effect of several factors. In order to study these factors, we may group them as follows:

1. Heat-transfer effects;
2. Steady-flow resistance;
3. Pulsation of the intake gases.

Actually there is considerable overlapping. The problem of heat transfer to a fluid flowing over a body or through a channel is closely associated with the frictional drag. Also, referring to the charging process of an engine cylinder, the friction in the port results in a heating of the charge. Under conditions where pulsations are important, the steady-flow characteristics of the valve and port combination determine the damping term in the vibration equation.[1] The mass term in such an equation is also affected by the steady-flow characteristics of the valve. Both inertia and damping may be reduced by providing a freer passage for the incoming charge.

From all points of view, therefore, the steady-flow characteristics of the inlet valve are important. Improving the flow characteristics of a valve is tantamount to increasing the effective area of cross-section of the inlet passage. Such an improvement, or increase, reduces the heating of the charge, decreases the resistance to steady flow, and increases the natural frequency of the intake gases considered as a vibrating system, as well as decreasing the damping.

It is at high engine speeds, where frictional drag and

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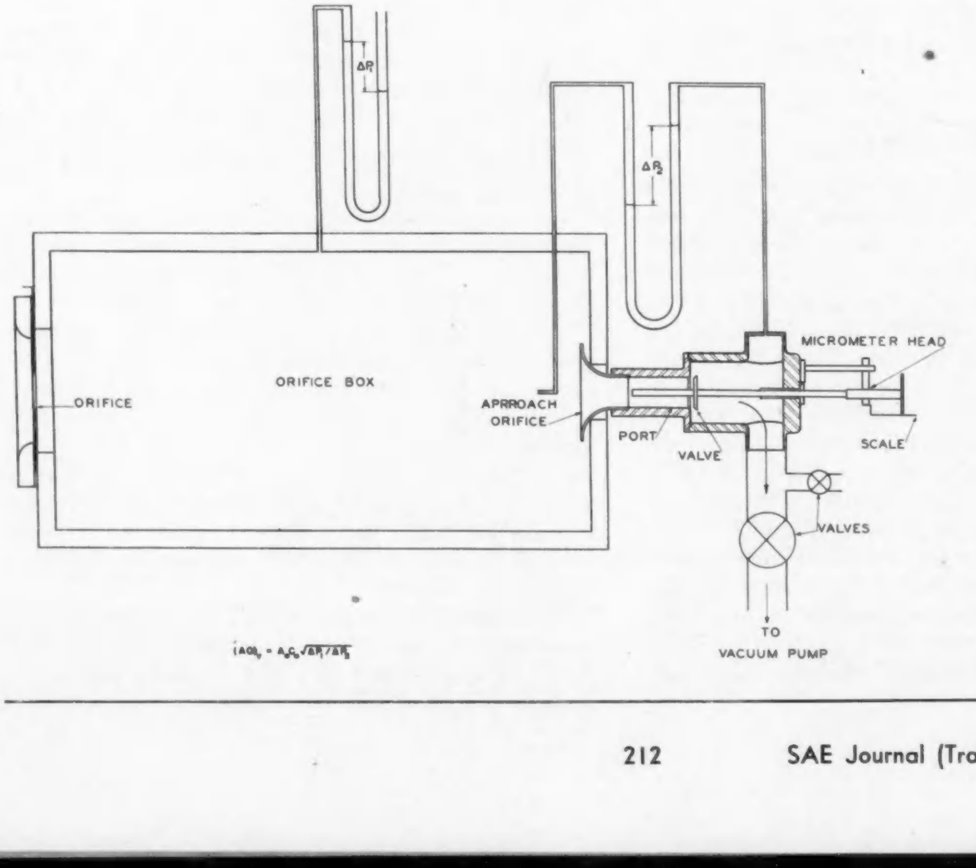
[1] See NACA Technical Note No. 675, 1938: "The Charging Process in a High-Speed Single-Cylinder, Four-Stroke Engine," by Blake Reynolds, Harry Schecter, and E. S. Taylor.

[2] See MIT Thesis, 1941: "Effect of Diameter Ratios on Flow through Inlet Valves of Internal-Combustion Engines," by Richard G. Falls and Stephen W. James.



■ **Fig. 2 – Apparatus for determining the steady-flow characteristics of a valve and port combination**

Through Intake Valves

by G. B. WOOD, JR.,* D. U. HUNTER,† E. S. TAYLOR,* and C. F. TAYLOR*

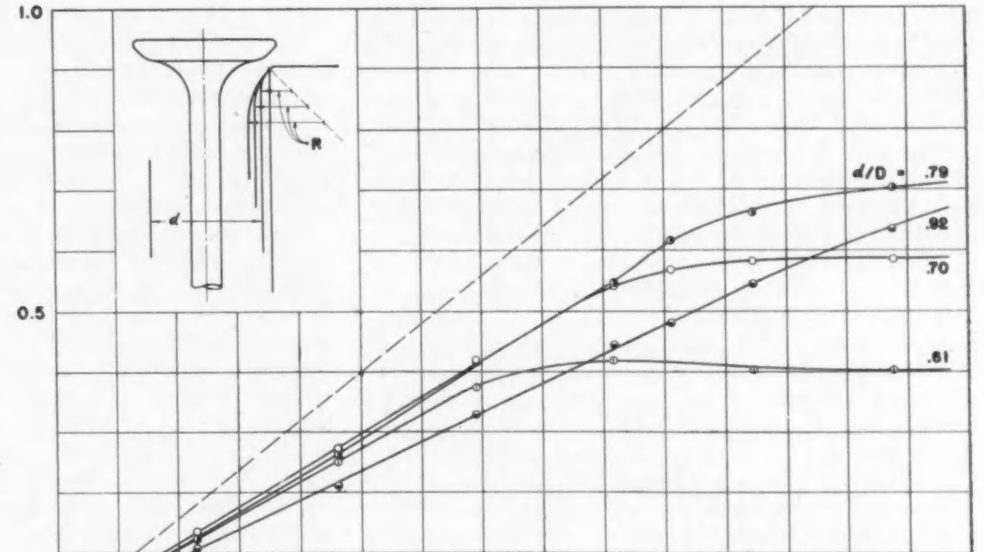
THE results of many flow tests made in the Sloan Laboratory at MIT on various valve and port combinations are presented in this paper.

A rational basis is developed for comparing valve and port combinations of different sizes and design, and theoretical factors are discussed. It is indicated that the conventional valve and port design may be greatly improved by comparatively simple modifications.

Most of the improvements appear to be due to the reduction of flow separation by eliminating sharp corners. A comparison of test conditions with actual operating conditions indicates how valve flow tests should be made and interpreted.

★ ★ ★

THE AUTHORS: G. B. WOOD, JR., gained his first intimate contact with internal-combustion engines in building and driving a racing car in 1933. He has been employed as a research and experimental engineer in the Sloan Laboratories at M.I.T. since December, 1938. He received his bachelor's degree in general engineering from M.I.T. in 1938, and collaborated with D. U. Hunter on a thesis on intake valves. D. U. HUNTER (J '40) has been working at the Lawrance Engineering and Research Corp. as an experimental engineer since 1941. Prior to that time he worked at Wright Aeronautical Corp. in a similar capacity. Mr. Hunter received a bachelor's degree in mechanical engineering from M.I.T. in 1938. He engaged in work on intake valves in preparing a college thesis, in collaboration with Author George B. Wood, Jr. E. S. TAYLOR (M '39), in addition to his duties as associate professor of aeronautics, in charge of the Sloan Automotive Engine Laboratory at M.I.T., is at present vice chairman of the NACA Committee on powerplants, and a consultant to the WPB. Prof. Taylor received the Reed Award in 1937 for the invention of the dynamic damper which solved the problem of torsional vibration in radial aircraft engines. He is also retained as a consultant by the Wright Aeronautical Corp. C. F. TAYLOR (M '19) was in charge of the U. S. Navy aeronautical engine laboratory during World War I. In 1919, Prof. Taylor resigned from active service with the rank of lieutenant in the Naval Reserve Force, and was appointed engineer in charge of the powerplant laboratory, engineering division, U. S. Air Service, McCook Field. From 1923 to 1926, he was with the Wright Aeronautical Corp., in charge of engine design and development. In the latter year he joined the faculty of M.I.T. as associate professor in aeronautical engineering. He was appointed professor of automotive engineering in 1933.



■ Fig. 3 – Intake-valve coefficients (Falls and James[2])

pulsation of flow become important, that the best correlation will be found between valve coefficients and air capacity, and hence, between valve coefficients and power output.

■ Testing Method and Basis for Coefficients

In general, tests of intake systems are run with only a small pressure drop across the apparatus. Under these conditions, air acts as an incompressible fluid, and we may use the simple hydraulics equation to define a discharge coefficient.

$$M = AC\sqrt{2\rho\,\Delta P}$$

M = mass flow
A = area
C = orifice coefficient
ρ = mass density of the fluid
ΔP = pressure drop (force units)

To study the flow characteristics of a valve and port a vacuum pump may be used to draw air through the combination. Where ρ and ΔP are fixed, the mass flow (M) is determined by the effective area (AC). For developing the design of an intake system on a particular engine this effective area AC is probably a good number to use to measure any improvements made in the design. However, it has the serious limitation that it does not reveal what stage of development has been reached.

To do this a characteristic area must be selected upon which a coefficient can be based. This coefficient may also be thought of as the ratio of the effective area of the valve to the characteristic area chosen as a base. This coefficient or ratio may then be employed to compare valve and port combinations from engines of different sizes and designs.

If a study were being made of an engine or cylinder, a good characteristic area would be the piston area or, perhaps even better, the 2/3 power of the piston displacement. However, for this paper, with the emphasis more directly on valve and port design, the circular area of the outside diameter of the valve head is taken as the base area. In many designs this diameter is the limiting dimension on the size of valve and port which can be used. Furthermore, it can be measured readily and does not change as the valve is lapped or pounds into its seat.

With the outside diameter of the valve fixed, it is possible to predict what a theoretically perfect valve would do (see Fig. 1). The port and valve head are assumed to be of the same diameter. It is also assumed that there is no friction between the walls of the channel and the flowing fluid, and that the flow does not separate from the walls.

The area of the discharge annulus of such a valve is given by the relation $A = \pi LD$, which becomes equal to the circular area of the valve head when $L/D = 0.25$. Curves showing valve coefficients are plotted against lift/diameter, with the basic curve shown for reference.

The apparatus required for determining the steady-flow characteristics of a valve and port combination is simple and inexpensive. Where not immediately available, it can be "cobbled up" quite conveniently. A block diagram of a typical layout is shown in Fig. 2. This apparatus will be similar whether it is to be used to study the flow characteristics of some simple part of a valve and port combination or for a complete engine intake system.

The calculations connected with such tests are also very simple. It is obvious, but nevertheless worth mentioning, that, when we use an orifice type of air meter in series with a valve and port combination, the mass flow of air through both of them will be the same. We may thus equate the expressions for this mass flow of air and obtain a simple formula for calculating valve coefficients as follows:

$$M = A_o C_o\sqrt{2\rho_1\,\Delta P_1} = A_v C_v\sqrt{2\rho_2\,\Delta P_2}$$

M = mass flow
A_o = area of orifice
A_v = basic area of valve
C_o = coefficient of orifice
C_v = coefficient of valve
ρ_1 = mass density of air before orifice
ρ_2 = mass density of air before valve
ΔP_1 = pressure drop across orifice
ΔP_2 = pressure drop across valve

Where $\Delta P_1 << P_1$

and $\rho_1 = \rho_2$ very nearly

$$C_v = \frac{A_o C_o}{A_v}\sqrt{\frac{\Delta P_1}{\Delta P_2}}$$

■ Test Results and Design Features

The logical place to start in developing a good intake port design is with an elementary valve and seat, eliminating the complicating factors of port elbow, cylinder-wall obstruction, and the like.

Probably the most important feature of a valve and seat combination designed to give high coefficients is the rounding of the corners immediately upstream from the throat or constriction between the valve and its seat. Many engines today have sharp corners at these locations. Even a slight breaking of the corners in the port and under the valve head will make an appreciable difference in the flow capacity, probably because it reduces the separation of the flow from the walls of the channel. Just how big the radius should be at these places depends to some extent on the design of the remainder of the port.

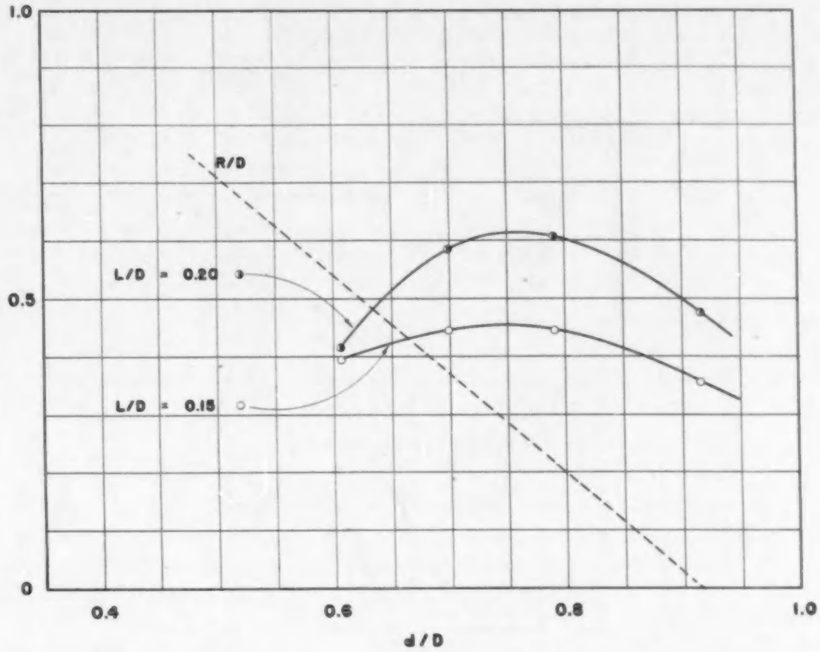
Using a cylindrical port and a smoothly faired valve, Falls and James[2] investigated the effect of varying the ratio of port diameter to valve diameter. Curves for four different ratios are shown in Fig. 3. In each case the radius joining the port to the seat is determined by the diameter of the port and of the inner edge of the seat. That is, the fairing was made tangent to both seat and port. The seat width was held constant at 0.176 in. (0.075 × valve head diameter) measured on the diagonal. A cross-plot of the results is shown in Fig. 4. Here C is plotted against port diameter divided by valve head diameter. The port to seat radius is also shown. With this particular valve and seat, it appears that the port diameter for maximum flow should be about ¾ of the valve-head diameter or, in other words, that the port to seat radius should be 0.2 times the valve-head diameter. On the other hand, it is quite possible that a port fillet radius between 0.2 times the valve head diameter and zero would have better flow characteristics than any of the ports tested. A considerable gap is left in the data between these two values.

A convenient method for developing valve and port designs was used by D. U. Hunter.[3] Starting with an elementary valve and seat, he built up various shapes with plasticene. Fairings of this material were added in the port, in the "cylinder" just beyond the port, under the

[3] See MIT Thesis, 1938: "The Effect of Valve and Port Shapes on Air Flow through Intake Valves," by D. U. Hunter.

valve head, and on the rim of the valve. One combination of these modifications is shown in Fig. 5 with some of the coefficients obtained. The heavy curve was obtained with port and "cylinder" modifications, and modification of the under-side of the valve head. Addition of the valve-head modification increases the flow, as shown by the dotted curve. This curve is based on the outside diameter of the original valve. The lighter solid curve shows the coefficient of this valve based on the enlarged outside diameter. The dot and dash curve shows for comparison the coefficients obtained with the elementary valve and seat.

A rather unusual feature of this design is the small fillet at the junction of the valve stem and valve head. Fig. 6 shows the effect of enlarging this fillet to proportions which might be more attractive as regards stresses in the valve.



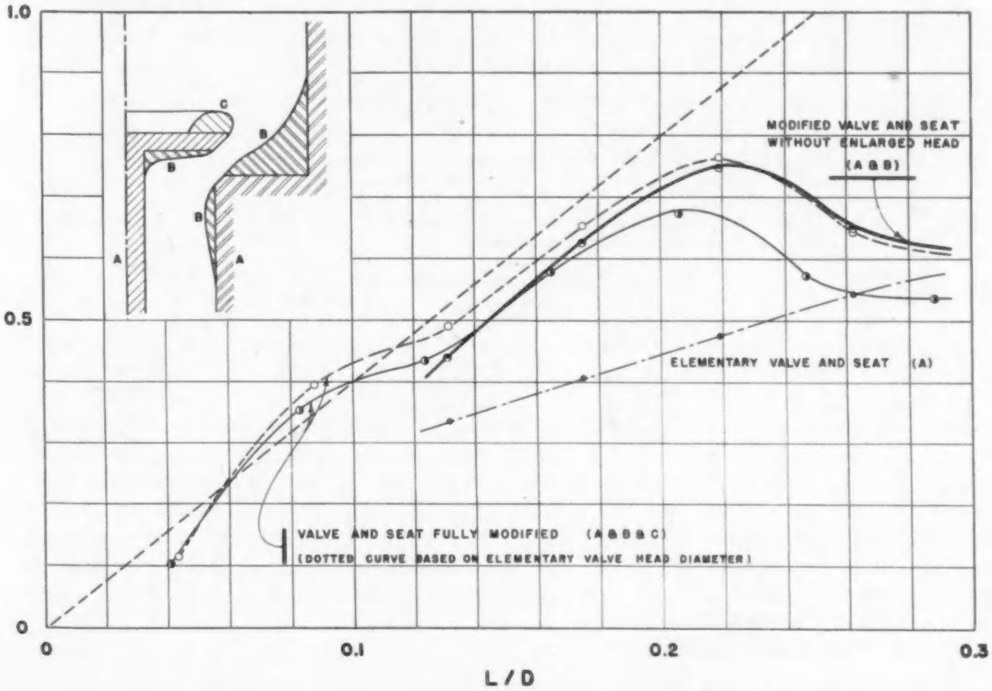
■ Fig. 4 – Cross-plot of results – intake-valve coefficients (Falls and James[2])

With valve and port combinations like those so far discussed it has been found that an appreciable pressure recovery may occur in the expanding portion of the passage between the valve and its seat. Although some of the experimental valves developed to achieve this end are impractical (see head enlargements in Figs. 5 and 14), it is possible to achieve an appreciable pressure recovery in what looks like a practical design.

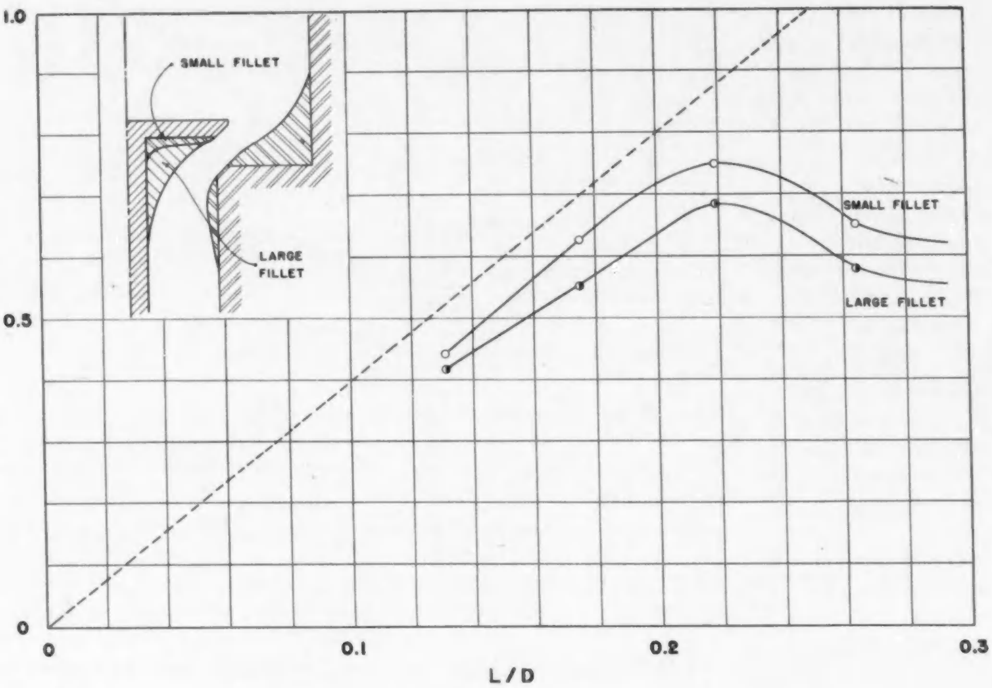
Fig. 7 shows the results of tests on a model of an L-head cylinder.[4] A well-developed intake valve was used in a straight port. The basic curve was made with the "cylinder head" removed, the other curve with the cylinder head installed and adjusted for best results. The difference reflects an improvement due to pressure recovery.

Fig. 8 shows a sketch of the apparatus and, at an arbitrarily selected valve lift, the effect of wall clearance and head clearance on the coefficients. It is quite apparent that, although the cylinder wall and head are often considered to obstruct the flow, they can be designed so as to assist it considerably. With this particular valve and port, best

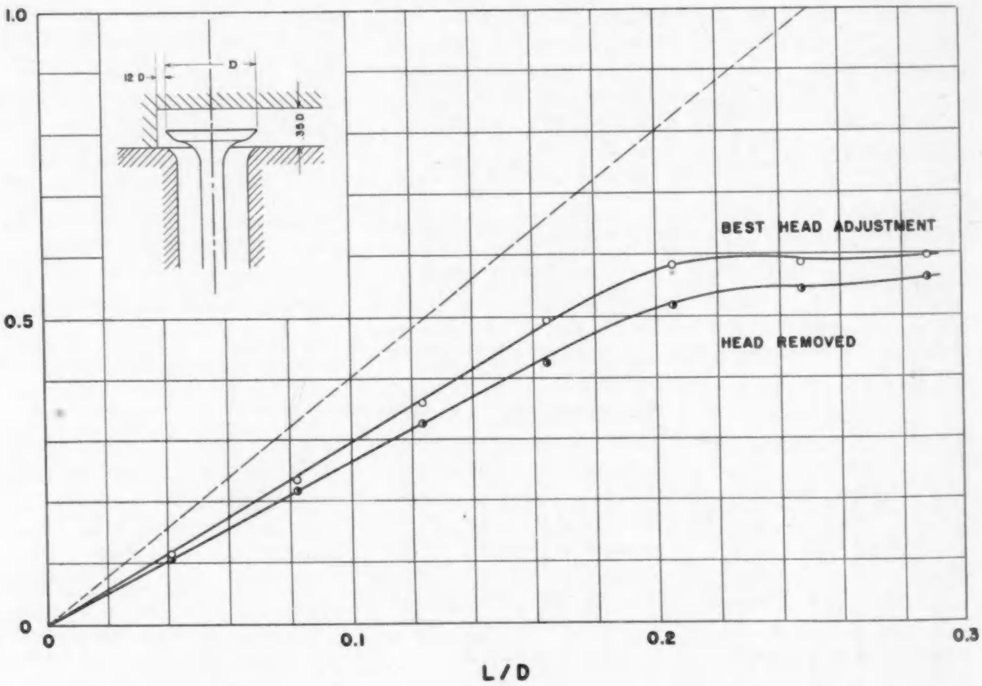
[4] See MIT Thesis, 1940: "Effect of Cylinder-Head Clearance on Flow Through an Inlet Valve of an L-Head Engine," by Wensley Barker, Jr., and James I. Thomas-Stahle.



■ Fig. 5 – Intake-valve coefficients (D. U. Hunter[3])



■ Fig. 6 – Effect of fillet on intake-valve coefficients (D. U. Hunter[3])



■ Fig. 7 – Intake-valve coefficients (Barker and Thomas-Stahle[4])

results were obtained with wall clearance about 1/8 and head clearance about 1/3 of the valve-head diameter.

The flow between a poppet intake valve and its seat is often disturbed considerably by the port elbow and by obstructions such as the valve guide and boss. The best elbow introduces some turbulence, and makes the flow unsymmetrical with respect to the valve axis.

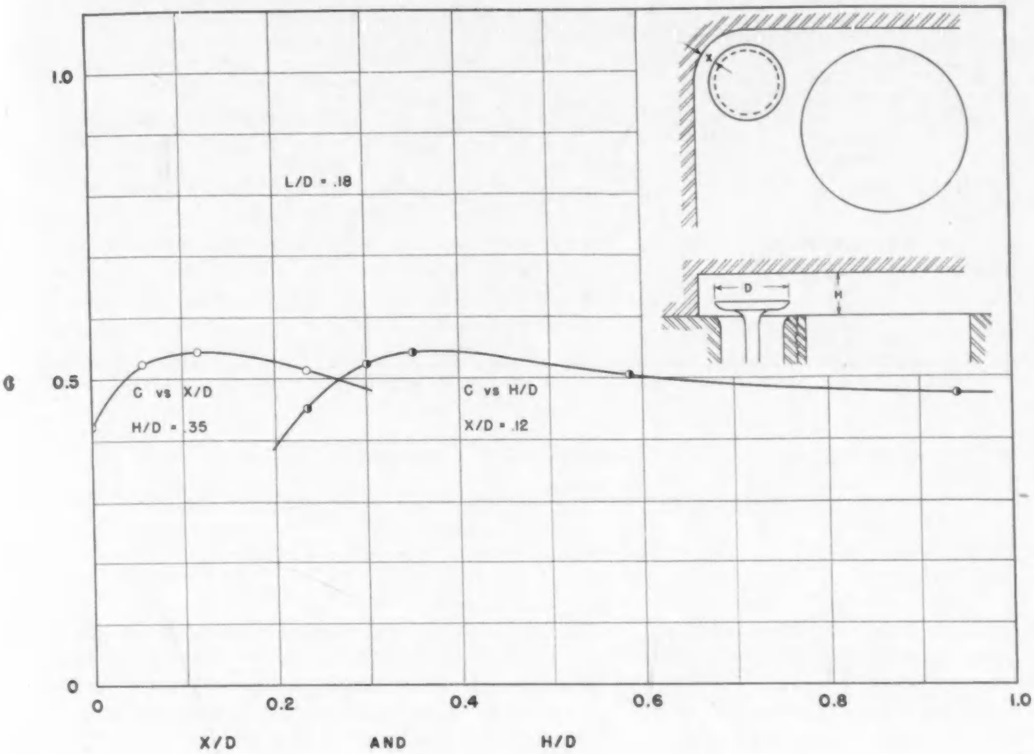
Under some conditions it appears that the elbow does not restrict the flow appreciably. Fig. 9 shows coefficients for a valve in a straight port and in an elbow port.[5] The elbow made no appreciable difference. However, it appears that the valve and seat design were not of the best, for the coefficient is generally low, and flattens off at a low value of L/D. It might be inferred that there was separation of the flow from the walls of the passage between the valve and its seat with the straight port as well as with the elbow port.

In the course of experiments in developing this elbow, several different radii of curvature were tested, fairing was installed behind the valve stem, and the effect of bulging

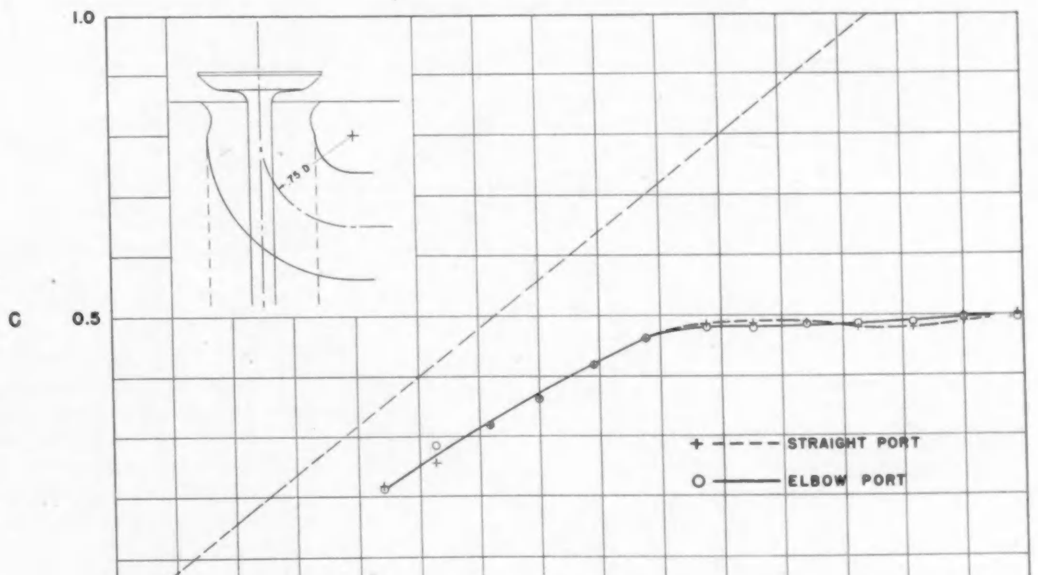
[5] See MIT Thesis, 1939: "The Effect of Inlet Port Elbows on Intake of the Internal-Combustion Engine," by George L. Estes, Jr., and James E. Hawkes.

the port at the bend was investigated. Fig. 10 shows coefficients plotted against bulge, $2B$, divided by valve-head diameter, and port radius divided by valve-head diameter. It appears that the bulge in the elbow should be about $0.2D$, or just about equal to the valve-stem diameter. This latter seems more likely to be the determining factor, but a complete investigation would require testing with several different sizes of valve stem.

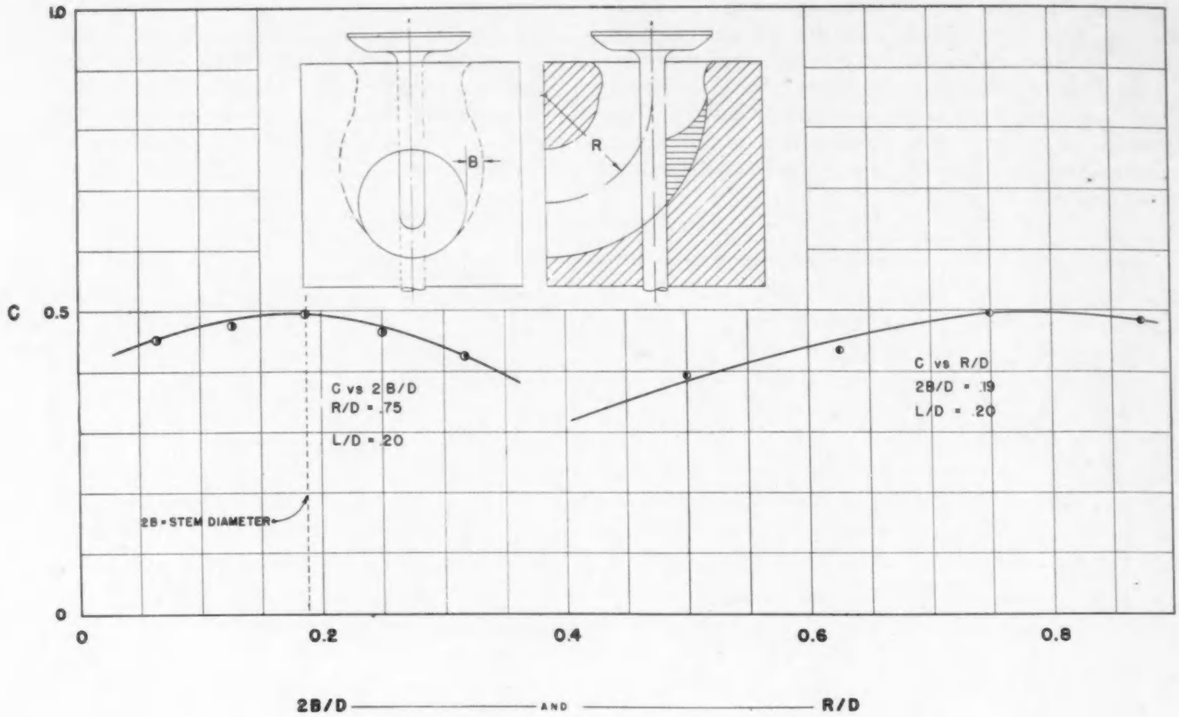
A generous radius of curvature of the elbow seems to be important in securing good coefficients. The reasons for this conclusion are obvious. What seems more surprising is the appearance of an optimum value at so small a radius. The offsetting factor which limits the port radius for best results may be the increased length of the port. No data are readily available to measure the effect of port length on air flow.



■ Fig. 8 – Apparatus and effect of wall clearance and head clearance – intake-valve coefficients (Barker and Thomas-Stahle[4])

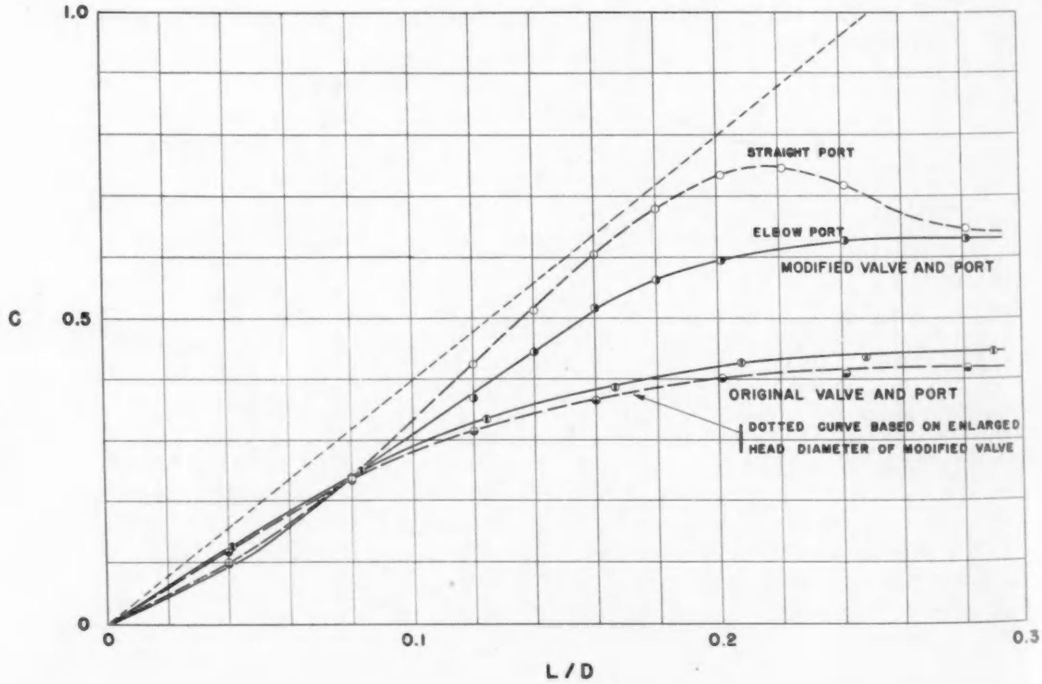


■ Fig. 9 – Intake-valve coefficients – straight port and elbow port (Estes and Hawkes[5])



■ Fig. 10 – Effect of radii, fairing, and bulging the port at the bend – intake-valve coefficients (Estes and Hawkes[5])

Coefficients obtained with an actual engine cylinder are shown in Fig. 11. This was an air-cooled airplane engine cylinder with one intake and one exhaust valve in the head. The solid curves show coefficients obtained before and after improvements in the valve and port. The modifications included changes in the intake port, valve, valve seat, and cylinder head. In the course of the work the valve head was enlarged, so the basis for the two solid curves is slightly different. The lower dotted curve shows the coefficient for the original valve, based on the outside diameter of the larger improved valve for comparison of the effective areas of the two.



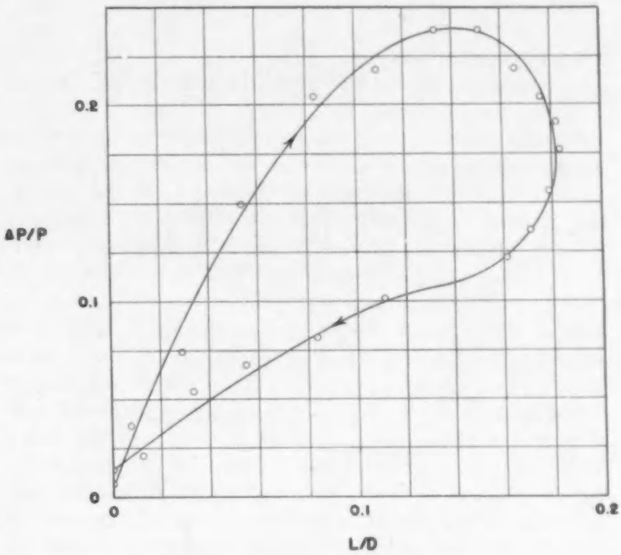
■ Fig. 11 – Intake-valve coefficients – results of work done at MIT for Ranger Engineering Corp.

The upper dotted curve shows the performance of the valve in a straight port with the improved valve seat and cylinder head. The considerable decrease in coefficient resulting from the addition of the port elbow is probably caused by two factors: The good design of the valve and seat make it sensitive to disturbances in the flow approaching it, and the elbow was necessarily somewhat cramped by space limitations.

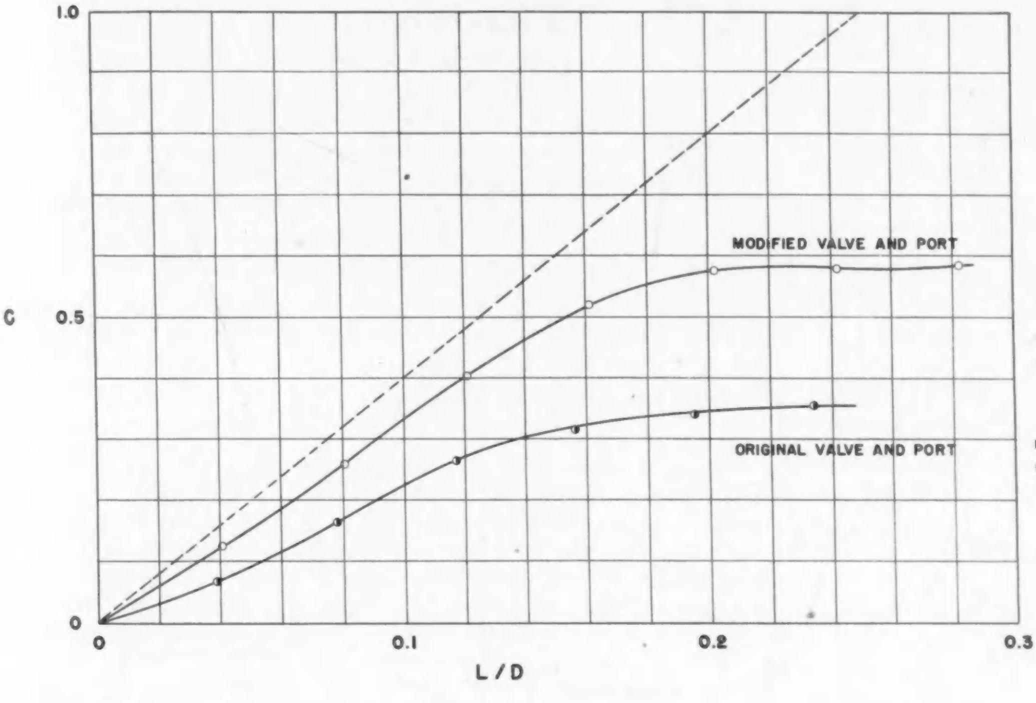
"Before" and "after" curves are shown in Fig. 12 for a much smaller aircooled cylinder of different design. Even greater improvement was made in this case, the final result being roughly the same as with the larger cylinder.

■ Pressure Drop in Actual Cylinder

As a guide to the order of magnitude of the pressure drop across the intake valve in an actual engine, Fig. 13 was prepared from NACA Technical Note No. 675 by Reynolds, Schecter, and Taylor.[1] This plot of $\Delta P/P$ versus L/D was made up by subtracting cylinder pressures from the atmospheric pressure. The cylinder pressure was read from indicator cards taken with the MIT high-speed engine indicator. Because of the mechanical limitations of the valve gear used, the piston speed of this engine was lower than that which would represent maximum power in a typical engine. At higher speeds the pressure drop ratio across the intake valve would be greater. On the other hand, the characteristics of the intake valve and port were not outstandingly good, and improvements might reduce the pressure drop considerably. The maximum $\Delta P/P$ of interest in practical cases is probably in the neighborhood of 0.3.



■ Fig. 13 – Pressure drop across intake valve (Reynolds, Schecter, and Taylor[1]) – piston speed 2380 fpm; intake pressure, atmospheric; intake pipe removed



■ Fig. 12 – Intake-valve coefficients – small aircooled cylinder

■ Effect of Pressure Drop

Most of the tests made at MIT to determine intake-valve flow characteristics have been made with a pressure drop of 10 in. of alcohol across the valve $\left(\frac{\Delta P}{P} = 0.02\right)$. This number was selected originally largely for the sake of convenience. The number is easily handled in the calculations, and it can be read accurately on a manometer. With this pressure drop ratio across the valve, the air velocity is high enough so that viscosity effects are negligible, but not so high that compressibility is important.

The flow of air between an intake valve and its seat is of similar nature to the flow through any channel which converges and then diverges, in other words, a venturi. The flow characteristics of such a system are determined by the effects of:

1. Viscous friction.
2. Separation of the flow from the walls of the channel.
3. Pressure recovery.

Viscous friction is probably not important where the pressure drop across the valve is not considerably less than 10 in. of alcohol. Separation of the flow from the walls of the channel tends to reduce the coefficients because it reduces the effective area of the passage. Pressure recovery tends to increase the coefficients because it increases the velocity at the throat with a given pressure drop across the valve. Where there is flow separation, it is difficult to achieve any large amount of pressure recovery, for the flow pattern is then not controlled by the shape of the passage.

In Fig. 14 a dotted line shows (plotted against the ratio of overall pressure drop to upstream pressure) the orifice coefficient that a perfect venturi would have if the calculations were based on the upstream pressure, the downstream pressure, and the throat area. The venturi was assumed to operate with no friction, no flow separation, and 100% pressure recovery. Coefficients for three different valve and seat designs are also shown plotted against pressure drop ratio. In order to make these coefficients comparable with that for the venturi, they are not based on the valve-head area but on the throat area, that is, the area of a conical frustum extending from the valve to its seat. This area was computed as follows:

$$A_t = A_s\,[2\sqrt{2}\;L/D + \sqrt{2}\;(L/D)^2]$$

where:

A_t = throat area

$A_s = \frac{\pi}{4} d^2$

L = valve lift

d = diameter at the inner edge of contact between valve and seat.

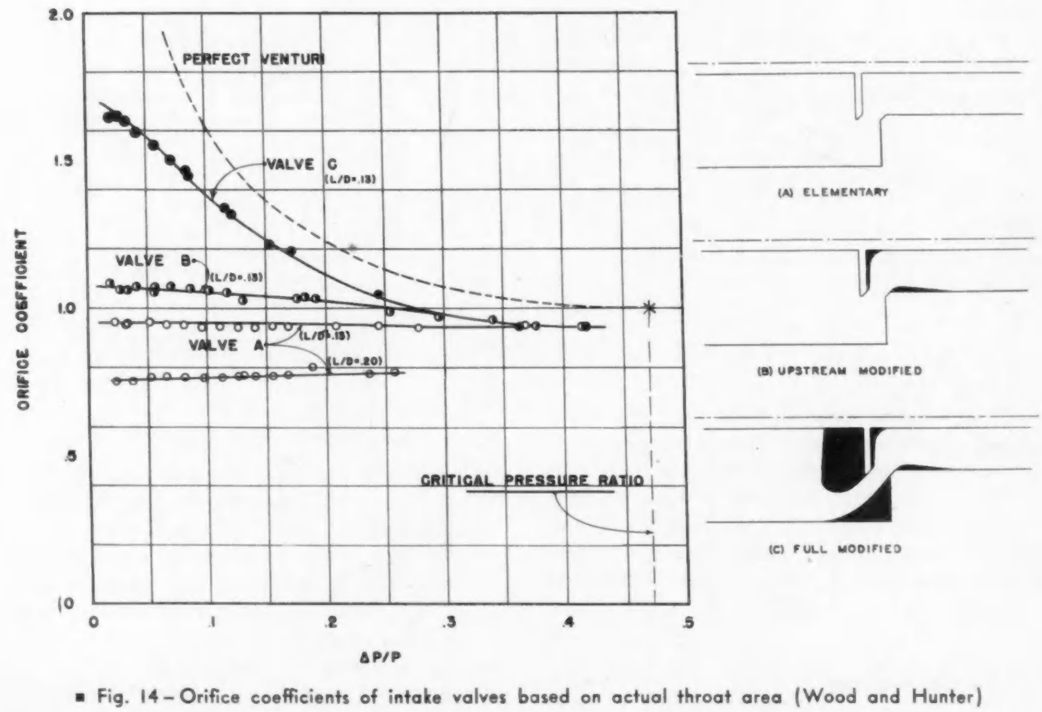
This method is not strictly accurate, being sometimes too large and sometimes too small, but is a good enough engineering approximation. Also, in calculating the valve coefficients, the simple hydraulic relations were modified to take into account the expansibility of the air.

From these intake valve coefficients based on the actual throat area it appears that the performance of intake valves may, under favorable conditions, approach the performance of a perfect venturi. Where the pressure drop ratio is low, the intake valve coefficient may be considerably greater than unity because of the pressure recovery in the diverging part of the passage. In such a case, the coefficient decreases rapidly when the pressure drop ratio is increased, as does the coefficient for a perfect venturi. Where the coefficient based on the actual throat area is not appreciably larger than one at low pressure drop ratio, it is not greatly affected by a change in the pressure drop ratio. This result leads to the conclusion that, except for valves of extreme design such as valve C of Fig. 14, tests at low pressure drop (10 in. alcohol) give a good indication of coefficients over the range of actual operation.

■ Effect of Valve Lift

The curves of C versus L/D show that, in many cases, the coefficient increases with increasing valve lift up to about $L/D = 0.2$, and thereafter remains substantially constant. At first glance it might be expected that nothing would be gained by having the maximum L/D greater than 0.2. That a higher lift can be used to advantage is obvious on second thought because, with higher lift, the

(Concluded on page 252)



■ Fig. 14 – Orifice coefficients of intake valves based on actual throat area (Wood and Hunter)

Characteristics of the VOLUTE SPRING

by BERNHARD STERNE
Experimental Engineer, Chrysler Corp.

THIS paper attempts to clarify the functioning of the volute spring and to eliminate the confusion which is at present connected with volute-spring computations. A number of contradictory formulas are currently in use, many of them usable only for one coil or for half a coil at a time.

The paper shows the relationship between the volute spring and other forms of coiled springs; it explains the similarities and dissimilarities of formulas for these spring forms, with particular emphasis on stress determination.

Because of the high stress invariably encountered in some part of a volute spring, it is clear that special consideration must be given to proper bulldozing and load-checking methods, and a set of specifications incorporating these methods is suggested.

In order to steer clear of excessive overstressing and the attendant spring settling, the stress reductions obtainable with partial tapering of the spring blade are discussed in detail.

As a proof for performance results which may be expected from a volute spring design, the life testing of the springs is considered indispensable, and an account is given of results in the past and of prospects in the future.

THE AUTHOR: BERNHARD STERNE (M '29) came to this country in 1925 from Germany after having received an early training in several of the leading continental automobile and machine-tool plants. In 1928, he joined the Chrysler Corp. as experimental engineer and was active for several years on aerodynamic and thermodynamic problems. For the past eight years his work has centered on car suspension. He is an experimental engineer with Chrysler.

THIS paper is concerned with a stepchild in the family of springs. The vastness of missing information is rivaled by the vagueness of existing information on the subject. Even expert authors of technical books and papers often don't know what they are talking about when they touch upon the volute spring.

The fault lies mainly with the confused terminology for the geometric properties which are involved. A typical description of the volute spring may be found in the "Century Dictionary and Encyclopedia," Revised Edition, New York 1911, Vol. X, page 6788, as follows:

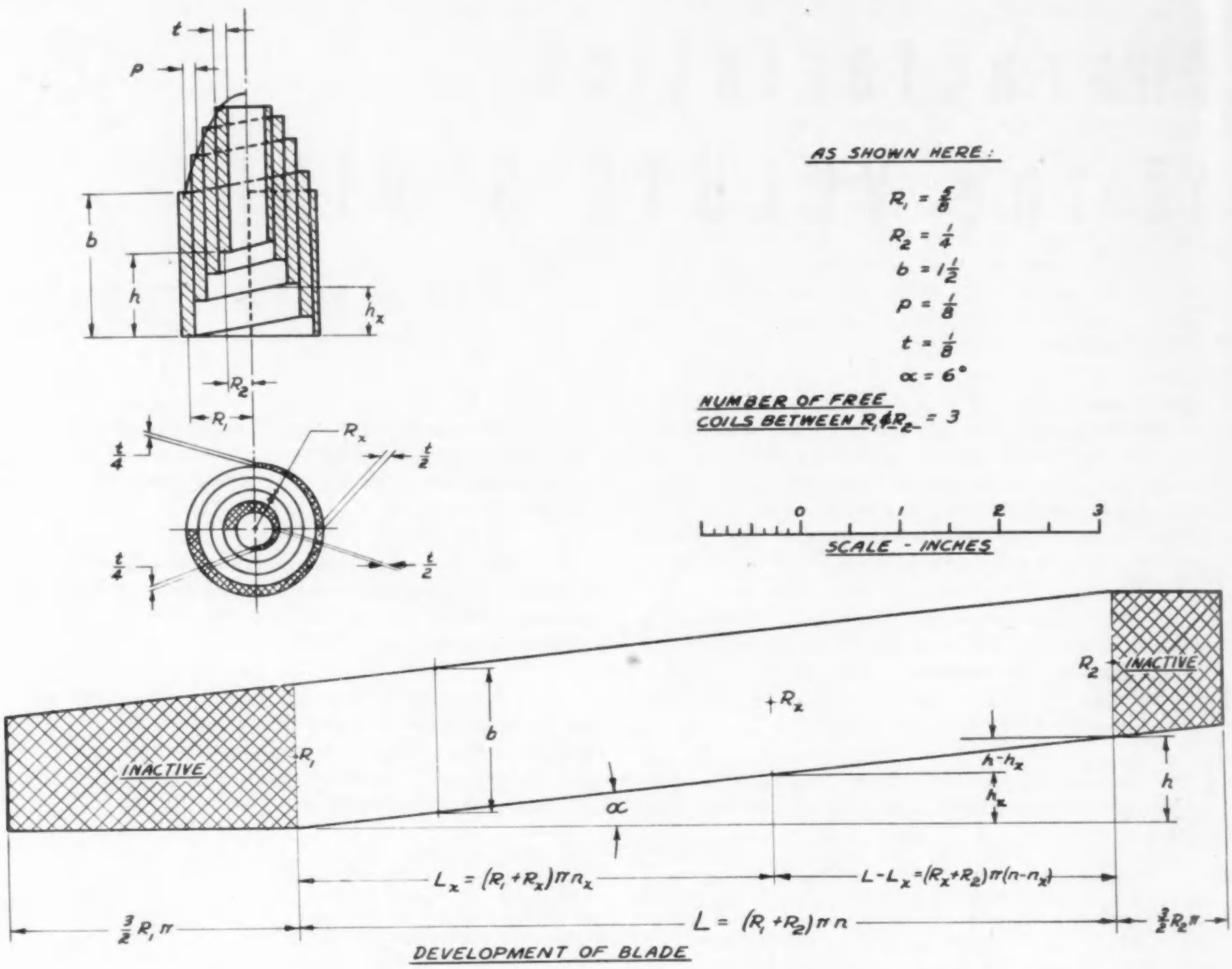
> "A spring consisting of a flat bar or ribbon, usually of steel, coiled in a helix somewhat in the form of a volute. It is commonly made in a conical form, so that the spring can be compressed in the direction of the axis about which it is coiled."

The engineer and the mathematician are used to considering the helix as a slanting line on the surface of a cylinder. On the other hand, the architect considers the helix distinctly as a flat spiral, and he knows the volute as a spiral scroll-shaped ornament. The volute spring has borrowed its name from this architectural scroll because of its outline in the plan view (Fig. 1), and its description as an architectural helix may therefore be excused. Still it remains confusing to the engineer to whom a "spring coiled in a helix" is a helical coil spring, and a helical coil spring is a spring coiled on a cylindrical arbor.

So much for the confusing first part of the definition in the Century Dictionary; but the second part is more than confusing – it is wrong. A spring "made in a conical form" implies an outline roughly in the form of a geometrical cone and a profile in the form of a triangle. There are conical coil springs answering that description (Fig. 2). When the active coils of such a spring are developed and projected into a plane, the slope of this ribbon will be found to vary continuously. In other words, when the bar for this spring is fed into the coiling machine the feed angle changes continuously, and the trailing end of the bar may have to be moved through a considerable arc during the coiling operation. When the bar has a round cross-section, it can adapt itself to this procedure fairly easily.

However, when the bar has a rectangular cross-section, and particularly when it consists of a blade which is comparatively small in radial thickness and comparatively large in axial width, then it can no longer adapt itself readily to a continuously changing feed angle during the coiling operation. The natural coiling method then will be to combine constant radial coil spacing with constant pitch angle. This results in a volute spring, where the slope of the developed coil is a constant, while the outline of the

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 12, 1942.]



■ Fig. 1 – Section, plan view, and development of volute spring with constant pitch angle α

spring is in the form of a paraboloid, and its profile is in the form of a parabola. One additional requirement to make this a true volute spring is, of course, that the radial spacing be at least equal to the thickness of the blade (or to the diameter of the round bar) so that, under maximum load, all coils will be nested within each other.

In all practical applications of conical springs and volute springs the coiling is not carried through to the apex of the enveloping cone or paraboloid. Consequently the outline of a practical conical spring is a truncated cone, and the outline of a practical volute spring is a truncated paraboloid. The volute spring is not a truncated conical spring as the Century Dictionary and other reference works would have one believe, and as even the punctilious Germans always refer to it.

The difference becomes important when the "bottoming" of the coils is considered. Let the various forms of coil springs be regarded as coiled from the bottom upward, or from the outside inward, contrary to the actual manufacturing process. The "bottoming" of any point x means that this point has reached the limit of its potential deflection and now becomes part of the inactive end coils. This potential deflection of point x will be called h_x as it is the height of point x from the bottom surface, minus the solid height of the coils up to this point x. The maximum deflection for the entire spring will be called h. In the case of the conical spring these deflections are proportional to the number of active coils:

$$\frac{h_x}{h} = \frac{n_x}{n}$$

On the other hand, in the case of the volute spring, the deflections are proportional to the developed length of the coils:

$$\frac{h_x}{h} = \frac{L_x}{L}$$

When the spring is deflected solid, the profile line will assume what may be called the "solid slope." In the case of the helical round wire spring, the solid slope is $\frac{\ }{2\pi R}$. In the case of the volute spring, the solid slope is zero since all coils are nested within each other and not on top of each other as in the helical spring.

When "bottoming" occurs at point x the profile line has been deformed until at that point it becomes parallel with the solid slope. In the case of the volute spring, the bottoming of point x takes place when the profile line at point x is at a right angle with the spring axis.

The initial part of the deflection in any coil spring takes

place under a constant rate. When the deflection has progressed far enough for the first active element in the spring to be bottomed and thus to be rendered inactive, then the process of reducing the number of active coils begins, with a transition from the original spring rate to higher and ever higher rates. The load under which this transition process begins will be called the transitional load ($P_{tr.}$).

When the helical spring is compressed, its pitch angle (or the slope of the developed profile line) is steadily reduced; but the profile line remains a straight line throughout the entire deflection, and all of its points reach the solid slope at one and the same time, with the result that the transitional load is also the maximum load. It should be said in parenthesis that this is only theoretically true. In practice some of the active coils near the end coils have a tendency to bottom and to increase the spring rate before the maximum load is reached.

The profile line of the conical spring is a curve with an obvious tendency towards gradual merging with the solid slope under progressive deflection, beginning at the point of the largest active coil radius R_1 (outer heel point) and eventually ending at the point of the smallest active coil radius R_2 (inner heel point).

The profile line of the volute spring is a straight line under no load, but it is deformed under load so that gradual merging with the solid slope will take place. Normally this merging will begin at R_1 and end at R_2 just as in the case of the conical spring; but there are important exceptions which will be discussed later.

These explanations of "bottoming" are a preliminary to considering the load P_x which will cause bottoming at any point x with a coil radius R_x. From the load formulas in Fig. 2 it will be seen that the load P_x is inversely proportional to R_x^3 in the conical spring and to $R_x^2\ (R_1 + R_2)$ in the volute spring. Also in Fig. 2 are shown deflection formulas, first for loads P which are less than the transitional load, and then for loads P_x which are greater than $P_{tr.}$. Under these loads P_x the deflection is made up of two components: one component (f_x') indicates the deflection in the coils which have already been bottomed (these normally are the coils with radii between R_1 and R_x); this component is different for the conical and for the volute spring; the other component (f_x'') indicates the deflection in the coils which still remain active (normally the coils with radii between R_x and R_2); this component is identical for the conical and for the volute spring.

■ Advantages and Disadvantages

All these relationships are interesting but, for the designer and user of the volute spring, they are generally of only secondary importance. He does not usually need to worry about static deflection, about frequency of the sprung mass, or about similar values obtainable from the load-deflection curve. But he does or at least should concern himself most seriously with the load capacity of the spring, and therefore with the stresses in the spring. The chief advantages of the volute spring, compactness and simplicity, are badly offset by the major disadvantage of a very poor stress distribution and, at that, a stress distribution which is poor on two cumulative counts: first within the cross-section of the blade, and second throughout the length of the blade. Incidentally, the story on stresses applies to both conical and volute springs. Between them the magnitude of the stress values will differ only because of the differences in the loads which produce the stresses, and which are inversely proportional to R_x^3 in the conical spring and inversely proportional to $R_x^2\ (R_1 + R_2)$ in the volute spring.

All coil springs are subject to torsion stress (T_c) under the compression loads with which we are here dealing. Before considering the torsion stress in the coiled bar, it will be well to examine the torsion stress in the straight bar. The straight *round* bar in torsion has a maximum stress which is constant at all points of the bar surface, and which is dependent upon its section modulus. The straight *rectangular* bar, according to de Saint-Venant's theory which is still authoritative after 85 years, has a maximum stress which is concentrated at the middle of the longer surfaces of the rectangle. It is dependent upon this longer surface (b), upon the square of the shorter surface (t^2), and upon the stress factor η_2. This stress factor, in turn, is dependent upon the ratio $\frac{b}{t}$. From the stress peak at the middle of the longer surfaces the stress recedes toward a minimum in the corners and increases again to a secondary peak at the middle of the shorter surfaces. In order to obtain this secondary peak, the maximum stress is multiplied by the factor η_1 which is again dependent upon the ratio $\frac{b}{t}$. The unequal stress distribution reduces the efficiency of the rectangular section by at least 30% against that of the round section.

When the bar is coiled, a further stress concentration takes place at the points nearest the axis of coiling, due to the effect of the curvature. A good many theories and empirical formulas for a stress-concentration factor have been brought forth on this effect, both for round and for rectangular cross-sections. In this country the best known formulas are probably those of Wahl and of Vogt. In the computations on which the graphs in this paper are based the Wahl formula has been employed.

When the spring is made of rectangular wire, the wire may be wound "flat" or "on edge." These two terms are employed here in a sense which is indicated in Fig. 2, and which may not be in agreement with the usage in some spring plants. If the wire is wound "flat," the longer surfaces are not subject to the curvature effect, and the maximum stress remains unchanged from that in the straight bar. The secondary stress peak, however, must be multiplied with the stress concentration factor and may thus equal or even exceed the stress at the middle of the longer sides. This results in a somewhat more even stress distribution than in the straight rectangular bar, and the coil spring of flat wound rectangular wire therefore finds favor with some spring experts.

In the volute spring, however, just the opposite takes place: This is the case of the rectangular wire wound "on edge"; the maximum stress at the middle of the longer surface which is nearer to the axis of coiling must be further multiplied with the stress concentration factor, and the result is a more uneven stress distribution than in any of the other cases, and consequently a further reduction in the efficiency of the spring.

So much for the stress distribution within the *cross-section* of the blade. The stress variation throughout the *length* of the blade is connected with the gradual bottoming of the coils. As long as the transitional load has not been reached, the highest stress is always in the largest

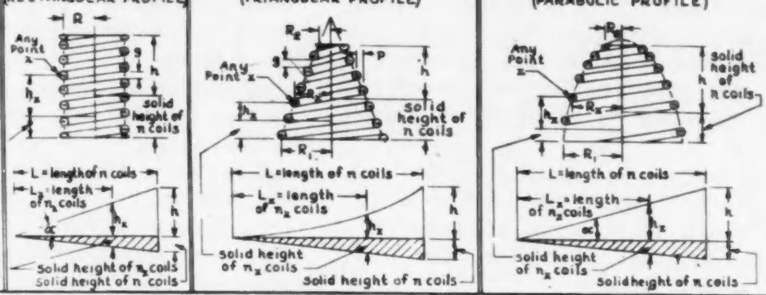
	HELICAL SPRING (RECTANGULAR PROFILE)	CONICAL SPRING (TRIANGULAR PROFILE)	VOLUTE SPRING (PARABOLIC PROFILE)

SPRING DIMENSIONS

DEVELOPED PROFILE OF SPRING

n = NO. OF ACTIVE COILS

h = MAXIMUM DEFLECTION OF SPRING



1. GEOMETRY

	Helical	Conical	Volute
p = Radial Coil Spacing	0	$\frac{R_1-R_2}{n}$	$\frac{R_1-R_2}{n}$
g = Vertical Gap Between Coils	$\frac{h}{n}$	$\frac{h}{n}$	Variable
n_x = No. of Coils to Point x	n_x	$\frac{R_1-R_x}{p} = \frac{R_1-R_x}{R_1-R_2}n$	$\frac{R_1-R_x}{p} = \frac{R_1-R_x}{R_1-R_2}n$
h_x Max. Deflection of Point x	h_x	$\frac{n_x}{n}h = \frac{R_1-R_x}{R_1-R_2}h$	$\frac{\pi n_x(R_1+R_x)}{\pi n(R_1+R_2)}h = \frac{R_1^2-R_x^2}{R_1^2-R_2^2}h$
(dh_x) = Differential of h_x	(dh_x)	$\frac{-h}{R_1-R_2}(dR_x)$	$\frac{-2hR_x}{R_1^2-R_2^2}(dR_x)$
L_x = Developed Length to Point x.	$\frac{h_x}{h}(2\pi nR)$	$\pi n_x(R_1+R_x) = \frac{\pi n(R_1^2-R_x^2)}{R_1-R_2}$	$\pi n_x(R_1+R_x) = \frac{\pi n(R_1^2-R_x^2)}{R_1-R_2}$
(dL_x) = Differential of L_x	$\frac{2\pi nR}{h}(dh_x)$	$\frac{-2\pi nP}{R_1-R_2}(dR_x)$	$\frac{-2\pi nR_x}{R_1-R_2}(dR_x)$
$\frac{}{(dL_x)}$ = Slope of Developed Coil at Point x	$\frac{h}{2\pi nR}$ (constant)	$\frac{\frac{-h}{R_1-R_2}}{\frac{-2\pi nR_x}{R_1-R_2}} = \frac{h}{2\pi nR_x}$ (variable)	$\frac{\frac{-2hR_x}{R_1^2-R_2^2}}{\frac{-2\pi nR_x}{R_1-R_2}} = \frac{h}{\pi n(R_1+R_2)}$ (Constant)

2. LOADS & DEFLECTIONS

	Helical	Conical	Volute
$\frac{(d\phi)}{(dL_x)}$ = Unit Angle of Twist at Point x (Load = P)	$\frac{M_t}{GJ} = \frac{PR}{GJ}$	$\frac{M_t}{GJ} = \frac{PR_x}{GJ}$	$\frac{M_t}{GJ} = \frac{PR_x}{GJ}$
$\frac{(df)}{(dL_x)}$ = Unit Deflection at Point x (Load = P)	$R\frac{(d\phi)}{(dL_x)} = \frac{PR^2}{GJ}$	$R_x\frac{(d\phi)}{(dL_x)} = \frac{PR_x^2}{GJ}$	$R_x\frac{(d\phi)}{(dL_x)} = \frac{PR_x^2}{GJ}$
Note	Point x is Bottomed when $\frac{(df)}{(dL_x)} = \frac{(dh_x)}{(dL_x)}$		
P_x = Load to cause solid bottoming at Point x when $\frac{(df)}{(dL_x)} = \frac{(dh_x)}{(dL_x)}$	$\frac{P_xR^2}{GJ} = \frac{h}{2\pi nR}$ $P_x = \frac{GJh}{2\pi nR^3}$	$\frac{P_xR_x^2}{GJ} = \frac{h}{2\pi nR_x}$ $P_x = \frac{GJh}{2\pi nR_x^3}$	$\frac{P_xR_x^2}{GJ} = \frac{h}{\pi n(R_1+R_2)}$ $P_x = \frac{GJh}{\pi nR_x^2(R_1+R_2)}$
Note	For Round Wire, J = Polar Moment of Inertia = $\frac{\pi d^4}{32}$ For Rectangular Wire, J' is a comparable quantity $J' = \eta_3 bt^3$ where b = length of longer side, t = length of shorter side, and η_3 is the St Venant deflection coefficient dependent on the ratio b/t		

	Helical: For Round Wire	Helical: For Rectangular Wire	Conical: For Round Wire	Conical: For Rectangular Wire	Volute: For Round Wire	Volute: For Rectangular Wire
P_x	$\frac{Gd^4h}{64nR^3}$	$\frac{G\eta_3bt^3h}{2\pi nR^3}$	$\frac{Gd^4h}{64nR_x^3}$	$\frac{G\eta_3bt^3h}{2\pi nR_x^3}$	$\frac{Gd^4h}{32nR_x^2(R_1+R_2)}$	$\frac{G\eta_3bt^3h}{\pi nR_x^2(R_1+R_2)}$
Note	Any load that causes point x to "bottom" would cause the entire Spring to be compressed solid		P_{tr} = Transitional load when largest coil starts to "bottom" P_{max} = Maximum load when smallest coil has bottomed then $P_x = P_{tr}$ when $R_x = R_1$ and $P_x = P_{max}$ when $R_x = R_2$			
	$P_{tr} = P_{max} = P_x$ = Gd^4h / [illegible] R^3	$P_{tr} = P_{max} = P_x$ $= \frac{G\eta_3bt^3h}{2\pi nR^3}$	$P_{tr} = \frac{Gd^4h}{64nR_1^3}$ $P_{max} = \frac{Gd^4h}{64nR_2^3}$	$P_{tr} = \frac{G\eta_3bt^3h}{2\pi nR_1^3}$ $P_{max} = \frac{G\eta_3bt^3h}{2\pi nR_2^3}$	$P_{tr} = \frac{Gd^4h}{32nR_1^2(R_1+R_2)}$ $P_{max} = \frac{Gd^4h}{32nR_2^2(R_1+R_2)}$	$P_{tr} = \frac{G\eta_3bt^3h}{\pi nR_1^2(R_1+R_2)}$ $P_{max} = \frac{G\eta_3bt^3h}{\pi nR_2^2(R_1+R_2)}$
Note	P_x is constant		$P_xR_x^3$ is constant		$P_xR_x^2$ is constant	
Spring Deflection under load P (less than P_{tr}) $= f = \int(df) = \int\frac{(df)}{(dL_x)}(dL_x)$	$\int_{h_x=0}^{h_x=h}\frac{PR^2}{GJ}\times\frac{2\pi nR}{h}(dh_x) = \frac{2\pi nPR^3}{GJ}$		$\int_{R_x=R_1}^{R_x=R_2}\frac{PR_x^2}{GJ}\times\frac{-2\pi nR_x}{R_1-R_2}(dR_x) = \frac{-2\pi nP}{GJ(R_1-R_2)}\int_{R_1}^{R_2}R_x^3(dR_x)$ $= \frac{\pi nP(R_1^4-R_2^4)}{2GJ(R_1-R_2)} = \frac{\pi nP(R_1^2+R_2^2)(R_1+R_2)}{2GJ}$			
f	$\frac{64nPR^3}{d^4G}$	$\frac{2\pi nPR^3}{G\eta_3bt^3}$	$\frac{16nP(R_1^2+R_2^2)(R_1+R_2)}{d^4G}$	$\frac{\pi nP(R_1^2+R_2^2)(R_1+R_2)}{2G\eta_3bt^3}$	$\frac{16nP(R_1^2+R_2^2)(R_1+R_2)}{d^4G}$	$\frac{\pi nP(R_1^2+R_2^2)(R_1+R_2)}{2G\eta_3bt^3}$
Note	For loads greater than P_{tr} (P_{max}) no further deflection occurs because the deflection under load P_{tr} is h		When point x is bottomed, the total deflection is made up of two components: f'_x = Deflection necessary to bottom n_x coils (Between R_1 and R_x) which are thus rendered inactive f''_x = Deflection of coils between R_x and R_2 which remain active and which themselves constitute a truncated spring having $n-n_x$ coils: Then the largest active coil radius, formerly R_1, now becomes R_x			
Deflection under load P_x (Greater than P_{tr}) $= f_x = f'_x + f''_x$			$f'_x = h_x = \frac{n_x}{n}h = \frac{R_1-R_x}{R_1-R_2}h$ $f''_x = \frac{\pi(n-n_x)P_x(R_x^2+R_2^2)(R_x+R_2)}{2GJ}$		$f'_x = h_x = \frac{R_1^2-R_x^2}{R_1^2-R_2^2}h = \frac{n_x(R_x+R_1)}{n(R_1+R_2)}h$ $f''_x = \frac{\pi(n-n_x)P_x(R_x^2+R_2^2)(R_x+R_2)}{2GJ}$	
f_x	h		$\frac{R_1-R_x}{R_1-R_2}h + \frac{16(n-n_x)P_x(R_x^2+R_2^2)(R_x+R_2)}{d^4G}$	$\frac{R_1-R_x}{R_1-R_2}h + \frac{\pi(n-n_x)P_x(R_x^2+R_2^2)(R_x+R_2)}{2G\eta_3bt^3}$	$\frac{R_1^2-R_x^2}{R_1^2-R_2^2}h + \frac{16(n-n_x)P_x(R_x^2+R_2^2)(R_x+R_2)}{d^4G}$	$\frac{R_1^2-R_x^2}{R_1^2-R_2^2}h + \frac{\pi(n-n_x)P_x(R_x^2+R_2^2)(R_x+R_2)}{2G\eta_3bt^3}$

■ Fig. 2 – Comparative loads, deflections, and

active coil, or more exactly in the spring element which has the largest active coil radius (R_1). When the load exceeds $P_{tr.}$ and becomes P_x, the maximum stress in the spring still occurs at the largest active coil radius, and that is now the coil radius R_x for which P_x is the bottoming load.

A comparison of the load and stress formulas will yield the information that the maximum stress within the cross-section is inversely proportional to the coil radius because the stress is proportional to $P R_x$ and P itself is inversely proportional to R_x^2. Thus the stresses are generally higher in the smaller coils; this is accentuated by a secondary cause in that the stress-concentration factor also increases as the coil radius decreases. In a volute spring with the largest coil radius equal to three times the smallest coil radius, the maximum stress in a spring element at the smallest coil radius may easily be 3½ times the maximum stress in a spring element at the largest coil radius. This unequal stress distribution throughout the length of the bar reduces the efficiency of the volute spring (and of the conical spring) by at least 50% against that of a helical spring with comparable cross-section. The overall efficiency of the regular flat-bladed volute spring can therefore be at best 1/3 of the overall efficiency of a round wire helical spring.

For a complete understanding of the volute spring the derivation of the formulas for loads, deflections, and stresses is given herewith, also a graph for the Wahl stress concentration factor (Fig. 3) and a graph for the stress factors η_1 and η_2 and the load factor η_3 (Fig. 4) which are needed for all computations with bars of rectangular cross-section.

Load, Stress, and Deflection Formulas for Volute Springs with Constant Vertical Pitch Angle α

Symbols - (Corresponding to Fig. 1 with outer and inner inactive coils tapered as shown there):

P_x = load on spring

R_1 = radius of largest active coil (to centerline of blade) $= \dfrac{OD - p}{2}$

R_2 = radius of smallest active coil (to centerline of blade) $= \dfrac{ID + p}{2}$

R_x = radius of coil just bottoming due to load P_x

n = total number of active coils

n_x = number of coils between R_1 and R_x

h = maximum possible deflection (all coils solid) $= (R_1 + R_2)\pi n\,(\tan\alpha)$

h_x = deflection of coils between R_1 and R_x

f_x = total deflection due to load P_x

b = height (width) of spring blade = solid spring height

t = thickness of spring blade

p = radial turn spacing $= \dfrac{R_1 - R_2}{n} = \dfrac{R_1 - R_x}{n_x} = \dfrac{R_x - R_2}{n - n_x}$
$= t + \text{gap}$
$= \dfrac{OD - ID}{2n + 2}$

$OD_{max.}$ = largest outside diameter measured across from the outer end of blade $= 2R_1 + p$

$ID_{min.}$ = smallest inside diameter measured across from the inner end of blade $= 2R_2 - p$

G = modulus of elasticity in torsion (11,000,000 psi used for steel)

η_2 and η_3 = de Saint-Venant coefficients (see Fig. 4)

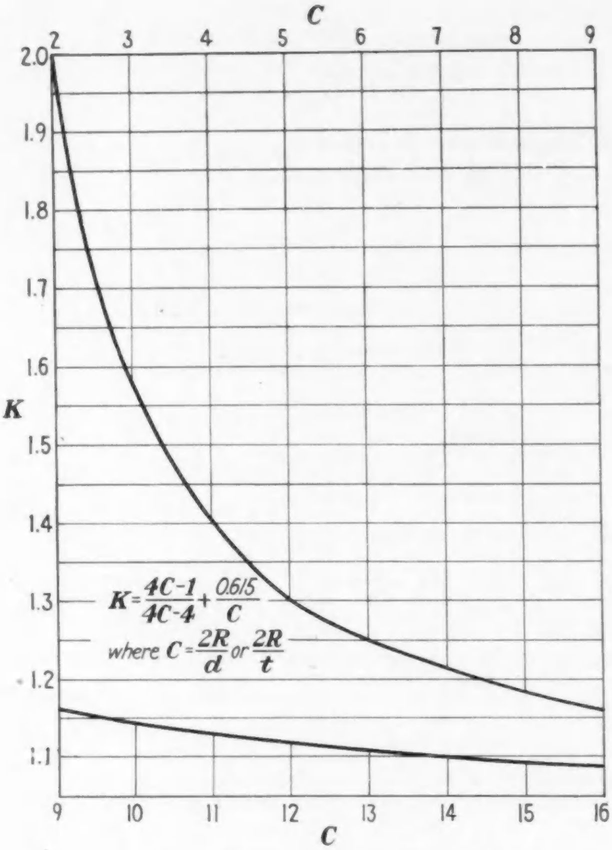
K = Wahl's stress concentration factor (see Fig. 3)

3. STRESSES

THE GENERAL FORMULA FOR MAXIMUM STRESS IN THESE SPRINGS IS $T_c = \frac{M_t y}{c} K$

Under a given load P the maximum stress within the cross section of the wire at any point Z $= T_{c_z} = \frac{PR_z y}{c} K_z$	For Round Wire: Maximum Stress Occurs at Point A	$c = J = \frac{\pi d^4}{32}$, $y = \frac{d}{2}$, K_z = Wahl factor or Vogt factor to allow for stress concentration effects caused by curvature; K_z is therefore dependent on the ratio $2R_z/d$				
	For Rectangular wire wound "on edge" (b>t): Maximum Stress Occurs at Point A	$c = \frac{\eta_2 bt^3}{2}$, $y = \frac{t}{2}$, K_z depends upon the ratio $2R_z/t$; η_2 = St. Venant's stress coefficient dependent upon the ratio b/t.				
	For Rectangular wire wound "flat" (b>t): Maximum Stress Occurs at Either Point A or at Point B	For stress at point A: $c = \frac{\eta_2 bt^3}{2}$, $y = \frac{t}{2}$, K = 1.0 (no curvature effect). For stress at point B: multiply stress at A by η_1 and also by the Wahl or Vogt factor K'_z (which is dependent upon the ratio $2R_z/b$. η_1 = St. Venant's factor relating the stress at B to the stress at A; it is dependent upon the ratio b/t.				
T_{c_z}	$\frac{16PR}{\pi d^3}K$	Wound "On Edge": $\frac{PR}{\eta_2 bt^2}K$; Wound "Flat" stress at A $= \frac{PR}{\eta_2 bt^2}$; stress at B $= \frac{\eta_1 PR}{\eta_2 bt^2}K'_z$	$\frac{16PR_z}{\pi d^3}K_z$	Wound "On Edge": $\frac{PR_z}{\eta_2 bt^2}K_z$; Wound "Flat": stress at A $= \frac{PR_z}{\eta_2 bt^2}$; stress at B $= \frac{\eta_1 PR_z}{\eta_2 bt^2}K'_z$	$\frac{16PR_z}{\pi d^3}K_z$	Wound "On Edge": $\frac{PR_z}{\eta_2 bt^2}K_z$; Wound "Flat": stress at A $= \frac{PR_z}{\eta_2 bt^2}$; stress at B $= \frac{\eta_1 PR_z}{\eta_2 bt^2}K'_z$
Under any given load the maximum stress in the spring which occurs at the largest active coil Radius R_x $= T_{c_x} = \frac{PR_x y}{c}K_x$	$T_{c_x} = T_{c_z}$		1) For load P less than P_{tr}: The above formulas apply with $R_z = R_x = R_1$ and $K_z = K_x = K_1$, which is dependent upon R_1. 2) For load P_x between P_{tr} and P_{max}: The above formulas apply with $P = P_x$ (= the load which will cause bottoming at radius R_x); $R_z = R_x$, and $K_z = K_x$ which is dependent upon R_x. 3) For load P_{max}: The above formulas apply with $P = P_{max}$, $R_z = R_2$, and $K_z = K_2$, which is dependent upon R_2.			
Note	For loads greater than P_{max}, stresses are not increased since the springs have been compressed solid by P_{max}.					

stresses in various types of coil springs



■ Fig. 3 – Wahl's stress concentration factor

Loads

The volute spring has a constant rate until such a value of load is reached that the largest coil starts bottoming, thus causing the spring rate to increase. The load at which this transition takes place will be designated as P_{tr}.

At any point on a volute spring with radius R_x an infinitesimal element consisting of (dx) coils will have *the same action as a similar element of a straight helical spring of the same radius R_x and the same pitch angle α.*

This equivalent helical spring element will have a deflection to compress the angle α to zero:

$(dh_x) = 2\pi R_x(dx)\tan\alpha$ from geometry of straight helix.

Referring to Fig. 5, for the element consisting of (dx) coils having a mean radius R_x:

length $(dL) = 2\pi R_x(dx)$

twisting moment $M_t = P_x R_x$

linear deflection $(df) = (dh_x) = R_x(d\phi)$

where $(d\phi)$ is the angle through which a force $P = P_x$ twists the element.

For torsional loading of a bar having rectangular cross-section:

$$(d\phi) = (dL)\theta = \frac{M_t}{G\eta_3 b\, t^3}(dL)$$

From Timoshenko & McCullough – "Elements of Strength of Materials," First Edition (1935) P. 78, Equation (49), which is derived from de Saint-Venant's theory.

where θ = angle of twist per unit of length.

Therefore:

$$(dh_x) = R_x(d\phi) = R_x\frac{P_x R_x}{G\eta_3 b\, t^3}2\pi R_x(dx)$$

$$(dh_x) = \frac{2\pi P_x R_x^3(dx)}{G\eta_3 b\, t^3}$$

Applying this to the equivalent elements of the helical and the volute spring:

$$(dh_x) = \frac{2\pi P_x R_x^3(dx)}{G\eta_3 b\, t^3} = 2\pi R_x\,(dx)\tan\alpha$$

Therefore the load P_x which will cause solid bottoming at radius R_x in the volute spring is:

$$P_x = \frac{G\eta_3 b\, t^3\tan\alpha}{R_x^2} = \frac{G\eta_3 b\, t^3 h}{\pi n R_x^2(R_1+R_2)}$$

When contact starts to occur at the large end, $R_x = R_1$, and

$$P_{tr.} = \frac{G\eta_3 b\, t^3\tan\alpha}{R_1^2} = \frac{G\eta_3 b\, t^3 h}{\pi n R_1^2(R_1+R_2)}$$

When contact point progresses to the small end, $R_x = R_2$, and

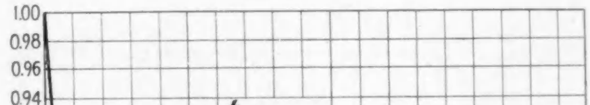
$$P_{max.} = \frac{G\eta_3 b\, t^3\tan\alpha}{R_2^2} = \frac{G\eta_3 b\, t^3 h}{\pi n R_2^2(R_1+R_2)} = P_{tr.}\times\frac{R_1^2}{R_2^2}$$

Stresses

Torsional stresses at any point on a volute spring having a radius R will be the same as for a helical spring of like radius and compression load P.

$$T_c = \frac{P\,R}{\eta_2 b\, t^2}\times K$$

Where K is the correction factor which takes account of the direct shear produced by the axial load and of stress concentration effects produced by the curvature of the wire. The factor K is therefore a function of the ratio $\frac{2R}{t} = C$ (for springs of round wire, $C = \frac{2R}{d}$).



■ Fig. 4 – De Saint-Venant's coefficients for stress (η_1 and η_2) and for deflection (η_3) of a rectangular bar in torsion

For K, either the Wahl or the Vogt factor may be used:

Wahl Factor:

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$

From ASME Research paper APM – 51-17.

Vogt Factor:

For tension springs: $$K = \frac{1.0875}{1-\frac{t}{2R}} = \frac{1.0875C}{C-1}$$

For compression springs: $$K = \frac{0.9875}{1-\frac{t}{2R}} = \frac{0.9875\,C}{C-1}$$

Both Vogt Factor formulas above from ASME Research paper RP – 58-14.

The largest free coil is always under the highest stress. Thus, for loads less than $P_{tr.}$, the maximum stress is at the largest radius R_1.

$$T_c = \frac{P\,R_1}{\eta_2 b\,t^2} K_1$$

When the load becomes $P_{tr.}$ the maximum stress is

$$T_{c_{tr.}} = \frac{P_{tr.}\,R_1}{\eta_2 b\,t^2} K_1 = \frac{\eta_3}{\eta_2}\,\frac{G \tan\alpha\,t}{R_1} K_1$$

For loads between $P_{tr.}$ and $P_{max.}$ the maximum stress is at radius R_x.

$$T_{c_x} = \frac{P_x R_x}{\eta_2 b\,t^2} K_x = \frac{\eta_3}{\eta_2}\,\frac{G \tan\alpha\,t}{R_x} K_x$$

When the load becomes $P_{max.}$, $R_x = R_2$, and

$$T_{c_{max.}} = \frac{P_{max.}\,R_2}{\eta_2 b\,t^2} K_2 = \frac{\eta_3}{\eta_2}\,\frac{G \tan\alpha\,t}{R_2} K_2$$

Deflections

The deflection of a volute spring with all coils active (load less than $P_{tr.}$) is derived as follows:

Referring to Fig. 5, an element consisting of (dx) coils is located x coils below the apex of the full volute spring. The radial pitch p, or horizontal spacing between adjacent coils, is constant and equal to $\frac{R}{n}$, where R is the maximum coil radius and n is the total number of coils.

The coil radius at point x is then

$$R_x = xp = \frac{xR}{n}$$

Under the action of a force P, the linear deflection of the element is

$$(df) = \frac{2\pi P\,R_x^3(dx)}{G\eta_3 b\,t^3} \text{ (see development of load formula)}$$

$$= \frac{2\pi P\,R^3x^3(dx)}{G\eta_3 b\,t^3 n^3}$$

For the entire spring under the load P, the deflection is

$$f = \int_{x=0}^{x=n} \frac{2\pi P\,R^3x^3(dx)}{G\eta_3 b\,t^3n^3} = \frac{2\pi P R^3\,\frac{n^4}{4}}{G\eta_3 b\,t^3n^3}$$

$$f = \frac{\pi n P\,R^3}{2G\eta_3 b\,t^3}$$

For a truncated volute, subtract the small top cone from the full cone, then

$$f = f_2 - f_1 = \frac{\pi P}{2G\eta_3 b\,t^3}(R_1^3 n_1 - R_2^3 n_2)$$

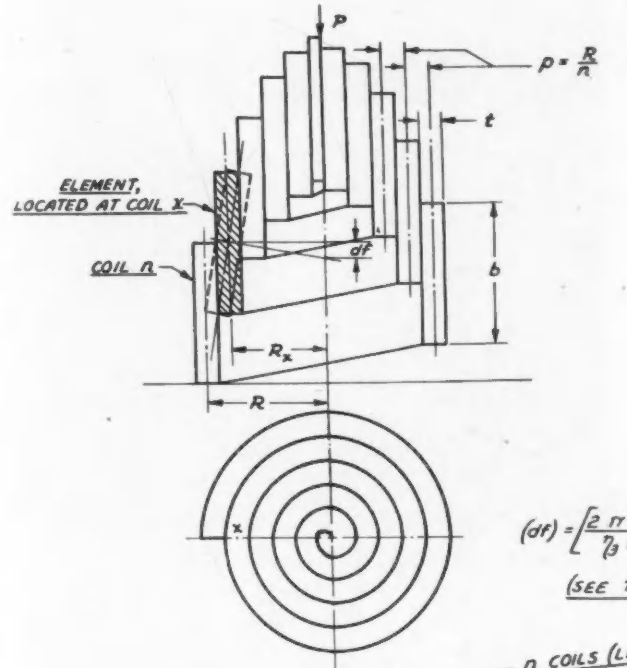
$$n_1 = \frac{R_1}{p} \qquad n_2 = \frac{R_2}{p}$$

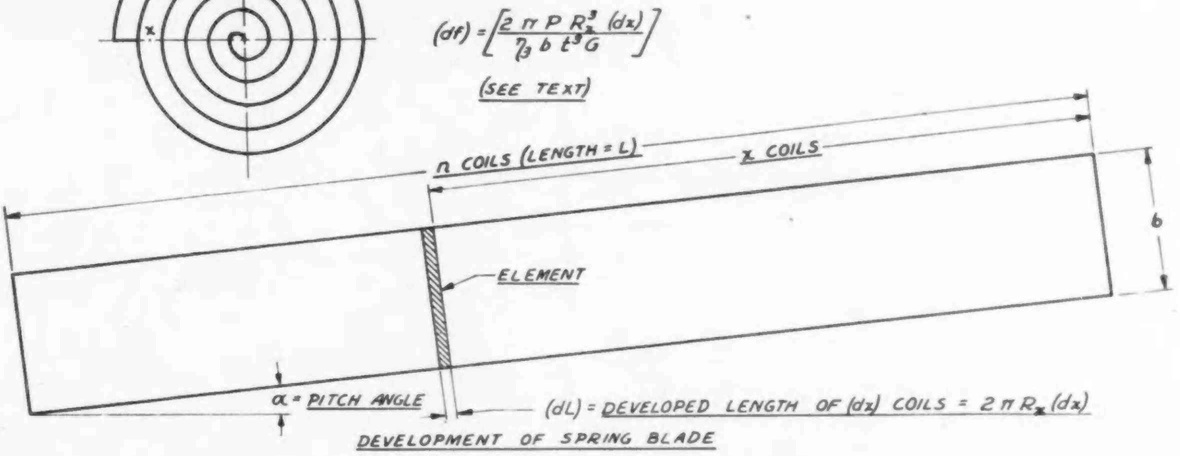
$$f = \frac{\pi P}{2G\eta_3 b\,t^3} \times \frac{R_1^4 - R_2^4}{p}$$

$$\text{but } n = n_1 - n_2 = \frac{R_1 - R_2}{p} \quad \text{or} \quad p = \frac{R_1 - R_2}{n}$$

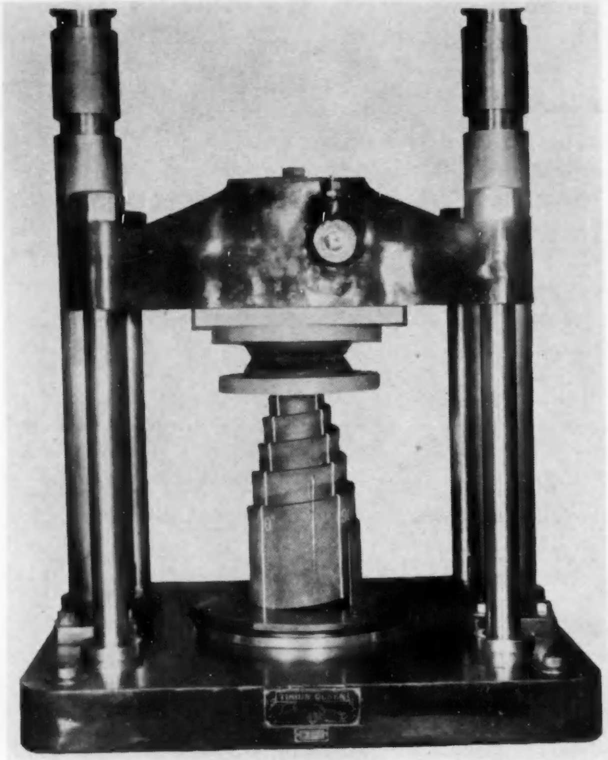
$$f = \frac{\pi n P}{2G\eta_3 b\,t^3} \times \frac{R_1^4 - R_2^4}{R_1 - R_2}$$

$$f = \frac{\pi n P}{2G\eta_3 b\,t^3}(R_1^2 + R_2^2)(R_1 + R_2)$$





■ **Fig. 5 – Deflection of an element of a spring coiled from a rectangular bar with constant pitch angle α (volute spring)**



■ Fig. 6 – Volute spring set up in checking machine with rubber biscuit

When P becomes $P_{tr.}$ and the largest coil starts to bottom, then

$$f_{tr.} = \frac{\pi n P_{tr.}}{2G\eta_3 b\, t^3}\,(R_1^2 + R_2^2)\,(R_1 + R_2) = h\,\frac{R_1^2 + R_2^2}{2R_1^2}$$

For loads between $P_{tr.}$ and $P_{max.}$ the deflection may be considered in two parts:

(a) the deflection which results in the bottoming of all coils between R_1 and R_x, and

(b) the additional deflection of the remaining (active) coils between R_x and R_2.

(a) $\dfrac{f_x'}{h} = \dfrac{\pi n_x (R_1 + R_x)}{\pi n (R_1 + R_2)}$ from geometry of volute spring, see Fig. 1.

$$\text{where} \quad n_x = \frac{R_1 - R_x}{p} = n\,\frac{R_1 - R_x}{R_1 - R_2}$$

$$f_x' = \frac{h\, n_x (R_1 + R_x)}{n(R_1 + R_2)} = h\,\frac{R_1^2 - R_x^2}{R_1^2 - R_2^2}\,.$$

$$\text{(b)}\ f_x'' = \frac{\pi(n - n_x)P_x}{2G\eta_3 b\, t^3}\,(R_x^2 + R_2^2)\,(R_x + R_2)$$

from deflection formula with all coils active, except that the number of remaining active coils (previously n)

$$= n - n_x = \frac{R_x - R_2}{R_1 - R_2}\,n$$

the large radius (previously R_1) $= R_x$

$$\text{the load} \quad P_x = \frac{G\eta_3 b\, t^3 (h - h_x)}{\pi(n - n_x)R_x^2(R_x + R_2)}$$

$$\frac{h - h_x}{\pi(n - n_x)\,(R_x + R_2)} = \tan\alpha = \frac{h}{\pi n(R_1 + R_2)}$$

$$P_x = \frac{G\eta_3 b t^3 h}{\pi n(R_1 + R_2)R_x^2}$$

$$f_x'' = \frac{\pi n(R_x - R_2)G\eta_3 b\, t^3 h\,(R_x^2 + R_2^2)\,(R_x + R_2)}{(R_1 - R_2)\pi n(R_1 + R_2)R_x^2\, 2G\eta_3 b\, t^3}$$

$$f_x'' = h\,\frac{(R_x^2 + R_2^2)\,(R_x^2 - R_2^2)}{2R_x^2(R_1^2 - R_2^2)}$$

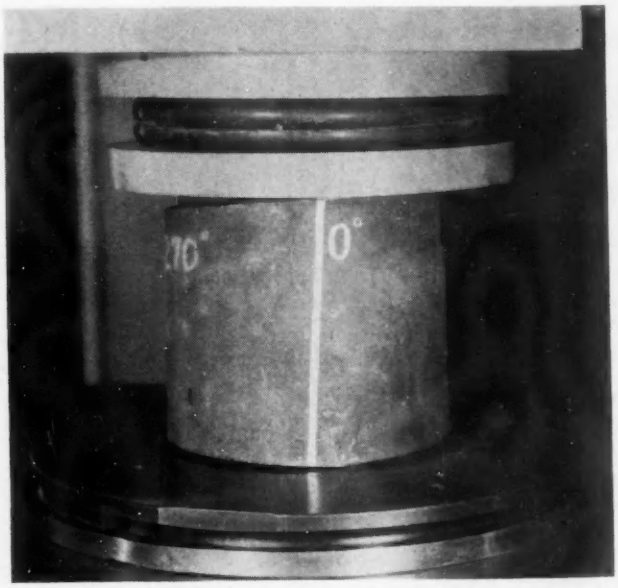
Therefore between $P_{tr.}$ and $P_{max.}$

$$f_x = f_x' + f_x''$$

$$f_x = h\,\frac{R_1^2 - R_x^2}{R_1^2 - R_2^2} + h\,\frac{(R_x^2 + R_2^2)\,(R_x^2 - R_2^2)}{2R_x^2(R_1^2 - R_2^2)}$$

At maximum load $n_x = n$, $R_x = R_2$, and $f_{x_{max.}} = h + 0 = h$

Loads larger than $P_{max.}$ are supported by the solid compressed spring, and there is no additional deflection or stress.

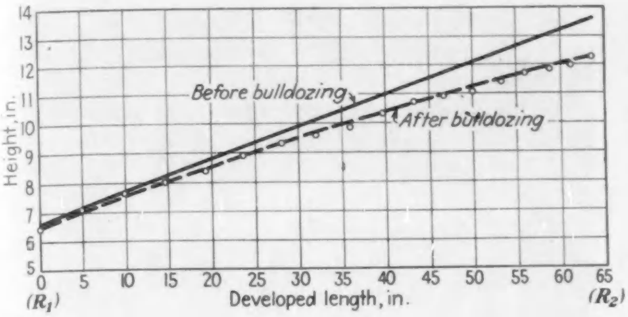


■ Fig. 7 – Volute spring and rubber biscuit compressed with 40,000-lb load

Table 1 – Experimental Check for Compression Load and Release Load on Volute Spring V-36

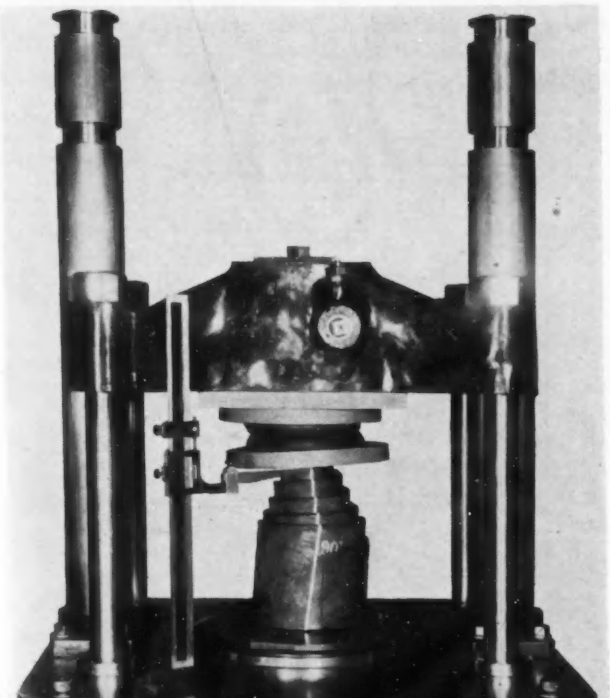
	Spring Rapped Before Each Load Reading				Spring Not Rapped During Checks			
	Compression Load	Release Load	Mean Load	Friction	Compression Load	Release Load	Mean Load	Friction
1. Check	8600	8075	8337	525	9300	7825	8562	1475
2. Check	8600	8075	8337	525	9375	7825	8600	1550
3. Check	8540	8050	8295	490	9525	7825	8675	1700
4. Check	8525	8050	8288	475	9600	7775	8687	1825
5. Check	8525	8050	8288	475	9500	7800	8650	1700
Average	8560	8060	8310	500	9460	7810	8635	1650
Range of Scatter	75	25	50	50	300	50	125	350

Note: Between each compression load and release load the spring was compressed to the solid height of 6½ in. with a load of 40,000 lb. The free height of the spring was checked after each release load and was found to vary only between 11.985 and 11.980 in.
All compression loads and release loads were checked at the static height of 8.500 in.



■ Fig. 8 – Slope of the developed blade of a volute spring before and after severe bulldozing

Probably in the majority of volute spring designs the enormous stress peak at the smallest coil is either overlooked or deliberately disregarded, generally in the assumption that the maximum load (and therefore the stress peak) will rarely if ever be reached. This assumption may be justified to a certain extent when the spring is adapted to such usage as in a railroad buffer. Not so, however, when the volute is used as a suspension spring. This application has recently received widespread attention, and it is well understood that in the operation of such vehicles over the most diversified terrain an occasional full bottoming of the suspension volute springs is bound to occur. If this bottoming is accompanied by stresses in excess of the elastic limit of the material and plastic deformation occurs, then a permanent set takes place. This causes the vehicle to lose



■ Fig. 9 – Volute spring under partial load

height, and the spring will bottom even more readily upon the next jar and will again take a permanent set. Eventually the spring will settle to the point where its total deflection becomes small and its maximum stress remains within the elastic limit. But its resistance to bottoming has then been sadly reduced, and the increasingly frequent stressing near the elastic limit will cause early fatigue failure.

Like any other spring, the volute spring will give proper service only after severe and probably repeated "bulldozing" with a load in excess of the maximum theoretical load. This systematic bulldozing procedure will cause a drastic permanent set and will remove the spring from the deflection range with stresses beyond the elastic limit. It will also cause considerable cold working in the spring which actually strengthens the spring material. The bulldozing operation may require the use of an elastic cushioning member if a loading device with positive mechanical drive is used. Fig. 6 shows the use of a rubber sandwich with a capacity of well above 40,000 lb. In the hydraulically operated Tinius Olsen machine, the rubber sandwich is not essential, but it will always help to protect any accurate loading machine from damage due to the sudden load increase when the spring is compressed solid. Fig. 7 shows the rubber sandwich with the spring solid under 40,000-lb load.

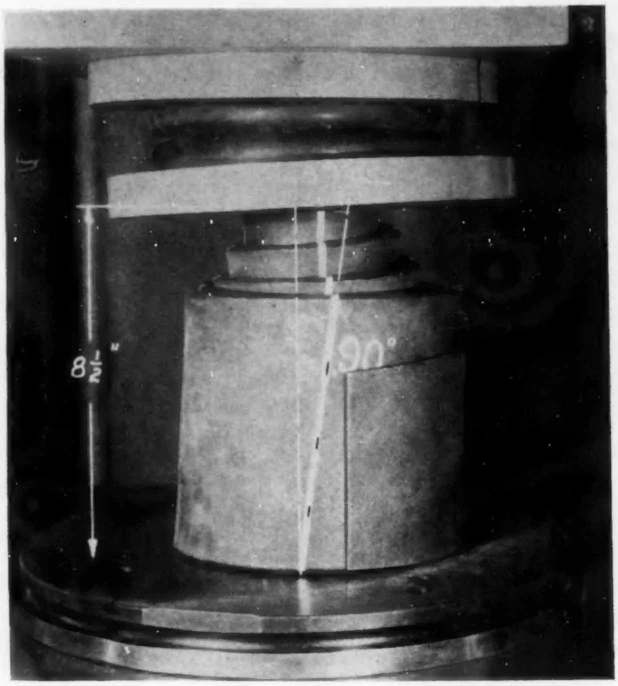
As a practical limit it is proposed to confine the bulldozing procedure to five consecutive applications of a load equal to 1¼ times the calculated solid capacity of the spring. A "Suggested Specification for Volute Springs" will be found in the Appendix[1] to the paper, and it contains this provision. It also provides that the spring be checked for working load at the specified loaded height (also called assembled height or static height) within 1 hr after the five bulldozing operations, in order to eliminate the more or less temporary "recovery" which takes place when the spring remains unloaded after bulldozing.

The working load is to be determined after thoroughly rapping the spring, because this is the method which conforms most closely to actual operating conditions, and it is also the only feasible method by which the result can be fairly checked and duplicated with reasonable accuracy. Table 1 shows the comparative results of five load checks after rapping the spring and five load checks without rapping. The five compression loads after rapping are scattered by 75 lb due to the 500 lb friction still in the spring despite the rapping. The five compression loads without rapping are scattered four times as much (300 lb). If the results of the so-called release load are added (that is the load when the spring is released from the solid height to the loaded height), and the mean of compression and release loads is established, then the scatter is reduced. However, this method is not as reliable as the use of the compression load after rapping, and it is considerably slower. Only for a check on the friction in the spring is the comparison of the unrapped compression and release loads of real value.

The bulldozing operations will cause the slope of the developed blade to be changed from a straight line to a more or less curved line. The shape of this curved line will depend a great deal upon the magnitude and the distribution of the stresses to which the spring has been subjected during the bulldozing.

In our laboratory we have carefully established the slope

[1] Copies of this Appendix are available on request to the Society of Automotive Engineers, 29 West 39th St., New York, N. Y.



■ Fig. 10 – Volute spring compressed to $8\frac{1}{2}$-in. height – 90-deg view

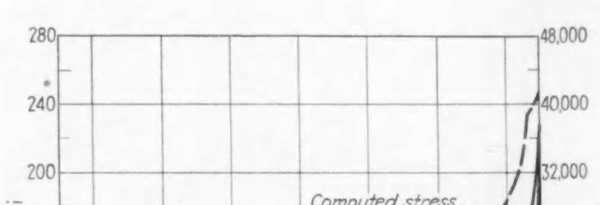
■ Fig. 11 – Volute spring compressed to $8\frac{1}{2}$-in. height – 270-deg view

of three reasonably identical production springs before and after a series of bulldozings, with fairly uniform results. The two slopes for one of these three springs are shown in Fig. 8. The upper line represents the slope of the spring just as it came out of production, and before it had undergone any loading after the coiling, except for a slight compression in the quenching fixture. The points for the lower line have been worked into a sweeping curve, but they may also be somewhat differently interpreted.

The slopes were checked at four points in each coil. For purposes of illustration these points were connected by vertical lines. Under increasing load these lines become more and more oblique as the distortion of the blade progresses. Fig. 9 shows the spring under partial load, with all coils still active, and consequently greater distortion in the outer coils, and the highest stress in the largest coil. In Fig. 10 the loading has progressed to a height of $8\frac{1}{2}$ in. About $1\frac{1}{2}$ outer coils have bottomed, and at a point 90 deg from the outer active coil radius R_1 the blade has been distorted by about 8 deg. Fig. 11 shows the spring under the same load from the opposite side, and it will be noted that, at 270 deg from R_1 the blade has been distorted by only about 4 deg. Quite likely there is a greater apparent distortion at 90 deg from R_1 because of the pivoting effect about the point with radius R_1 which lifts a part of the outer inactive coil from the spring seat. This effect may be plainly seen in the illustration. It will also be noted that the rubber biscuit has tilted to a certain angle. This is proof of the considerably eccentric loading in the spring. When the rubber biscuit is omitted the appearance of the spring under load does not change materially.

In Fig. 12 the load-deflection curve for one of the production springs is compared with the computed load-deflection curve for this spring. The difference between the two is a measure for the loss in capacity during bulldozing because of excessive stresses. Also shown is a computed maximum stress curve, which indicates the maximum stress in the spring corresponding to each load point in the loading curve. The stress curve climbs to a peak at 268,000 psi, and settling must obviously occur during the first loading since this stress peak is not only in excess of the elastic limit but even in excess of the ultimate strength of the material. The settling in this case has amounted to about $\frac{3}{4}$ in.

Thus we are confronted at every turn with the question



■ Fig. 12 – Comparison of computed and actual load-deflection curves for 7-in. OD production volute spring

of excessive stresses. If anything is to be done about them, it will be well to examine the stress formulas with a critical eye. The maximum stress may be expressed in this way:

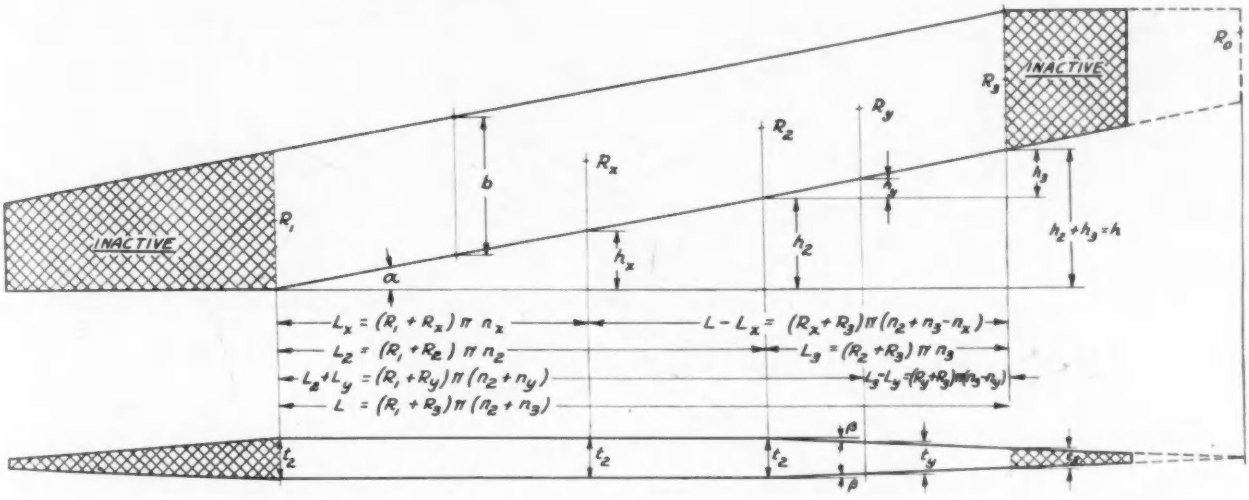
$$T_{c_{\text{max.}}} = \frac{n_3}{n_2} \frac{G \tan \alpha \, t}{R_2} K_2$$

$\tan \alpha$ is a measure of the potential deflection in the spring. This is the value which is reduced when settling takes place. As long as the spring is expected to be active throughout a given range of deflection, $\tan \alpha$ cannot be changed.

An increase in the inner coil radius R_2 would reduce the stress but, in a closely wound spring which is confined in a given space, R_2 cannot be increased, unless it were done by removing part of the inner coil, and this would increase $\tan \alpha$ and increase rather than decrease the maximum stress.

The only remaining factor in the formula is the blade thickness t, whereas the blade width b does not affect the stress at all. Actually, the reduction of the blade thickness is a practicable means of reducing the stress peak, and it should be more widely employed. To be sure, the reduction of t will also reduce the load capacity of the spring, but as long as the tapering is confined to the small inner coils, the performance of the spring will not be appreciably influenced in the static deflection range, and by virtue of the reduction of settling loss the overall performance will be improved.

Fig. 13 represents the developed blade of such a partly tapered spring. The symbols conform with those used in Fig. 1 except for the following details: R_2 now indicates the coil radius at which tapering begins, and the uniform blade thickness up to this point has been designated as t_2. The smallest active coil radius is now called R_3, and between R_2 and R_3 the thickness is tapered down from t_2 to t_3. The spring may be considered built up of two parts, the lower part having n_2 active coils between the coil radii R_1 and R_2, a maximum potential deflection h_2 and a uniform blade thickness t_2, while the upper part has n_3 active coils between the coil radii R_2 and R_3, a maximum potential deflection h_3 and a blade thickness varying from t_2 to t_3. The tapering continues into the inner inactive coil and may theoretically continue beyond the end of the blade



$$\tan\alpha = \frac{h_x}{L_x} = \frac{h_2}{L_2} = \frac{h_y}{L_y} = \frac{h_3}{L_3} = \frac{h}{L}$$

$$p = \frac{R_1-R_x}{n_x} = \frac{R_1-R_2}{n_2} = \frac{R_2-R_y}{n_y} = \frac{R_2-R_3}{n_3} = \frac{R_1-R_3}{n_2+n_3}$$

$$\tan\beta = \frac{t_2-t_3}{2L_3}$$

$$h_x = \pi \tan\alpha(n_2+n_3)\frac{R_1^2-R_x^2}{R_1-R_3} = h_2\frac{R_1^2-R_x^2}{R_1^2-R_2^2}$$

$$h_2 = \pi \tan\alpha(n_2+n_3)\frac{R_1^2-R_2^2}{R_1-R_3}$$

$$h_y = \pi \tan\alpha(n_2+n_3)\frac{R_2^2-R_y^2}{R_1-R_3} = h_3\frac{R_2^2-R_y^2}{R_2^2-R_3^2}$$

$$h_3 = \pi \tan\alpha(n_2+n_3)\frac{R_2^2-R_3^2}{R_1-R_3} = (h_2+h_3)\frac{R_2^2-R_3^2}{R_1^2-R_3^2}$$

$$\frac{n_x}{n_2+n_3} = \frac{R_1-R_x}{R_1-R_3} \qquad R_x = R_1 - \frac{n_x}{n_2+n_3}(R_1-R_3)$$

$$\frac{n_x}{n_2} = \frac{R_1-R_x}{R_1-R_2}$$

$$\frac{n_2}{n_2+n_3} = \frac{R_1-R_2}{R_1-R_3} \qquad R_2 = \sqrt{\frac{L_3(R_1-R_3)}{\pi(n_2+n_3)} + R_3^2}$$

$$\frac{n_y}{n_2+n_3} = \frac{R_2-R_y}{R_1-R_3} \qquad R_y = R_1 - \frac{n_2+n_y}{n_2+n_3}(R_1-R_3)$$

$$\frac{n_y}{n_3} = \frac{R_2-R_y}{R_2-R_3}$$

$$\frac{n_2+n_y}{n_2+n_3} = \frac{R_1-R_y}{R_1-R_3}$$

$$\frac{n_3}{n_2+n_3} = \frac{R_2-R_3}{R_1-R_3}$$

$$R_0 = \sqrt{\frac{t_2R_3^2 - t_3R_2^2}{t_2-t_3}}$$

$$t_y = \frac{R_y^2(t_2-t_3) + t_3R_2^2 - t_2R_3^2}{R_2^2-R_3^2} = t_2\frac{R_y^2-R_0^2}{R_2^2-R_0^2} = t_3\frac{R_y^2-R_0^2}{R_3^2-R_0^2} = (t_2-t_3)\frac{R_y^2-R_0^2}{R_2^2-R_3^2}$$

▪ Fig. 13 – Development of volute spring with constant pitch angle α, and with thickness tapered from t_2 to t_3 between R_2 and R_3

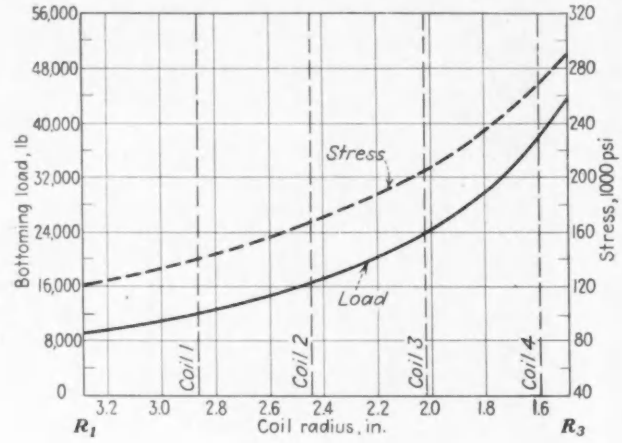


Fig. 14 – Load and stress versus coil radius

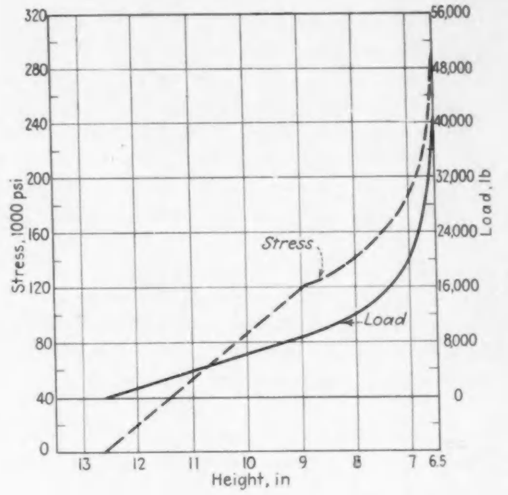


Fig. 15 – Load and stress versus height

■ Figs. 14 and 15 – Curves for a volute spring – 7-in. OD; 6.5 x 0.350-in. blade; 4¼ active coils; no taper in active coils

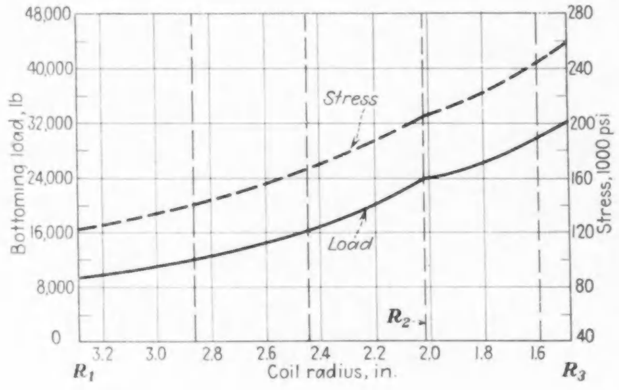


Fig. 16 – Load and stress versus coil radius

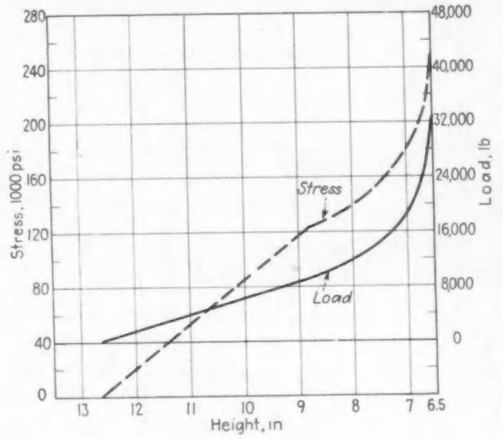


Fig. 17 – Load and stress versus height

■ Figs. 16 and 17 – Curves for a volute spring – 7-in. OD; 6½ x 0.350-in. blade; 4¼ active coils; blade tapered from end of third coil to a thickness of 0.300 in. at inner end

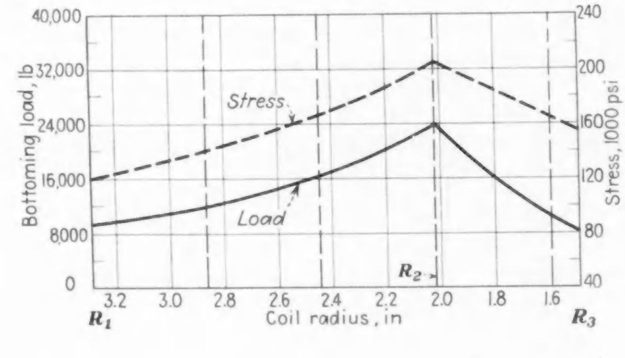


Fig. 18 – Load and stress versus coil radius

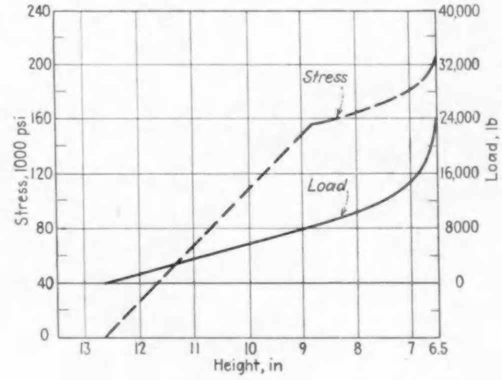


Fig. 19 – Load and stress versus height

■ Figs. 18 and 19 – Curves for a volute spring – 7-in. OD; 6½ x 0.350-in. blade; 4¼ active coils; blade tapered from end of third coil to a thickness of 0.125 in. at inner end

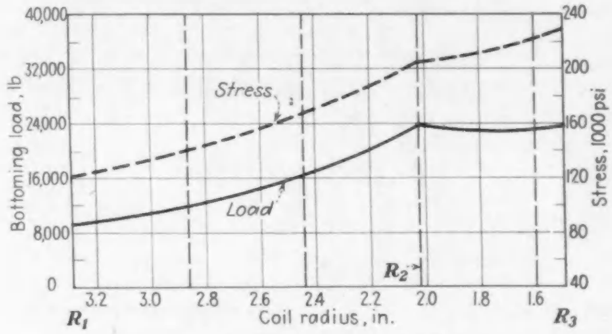


Fig. 20 – Load and stress versus coil radius

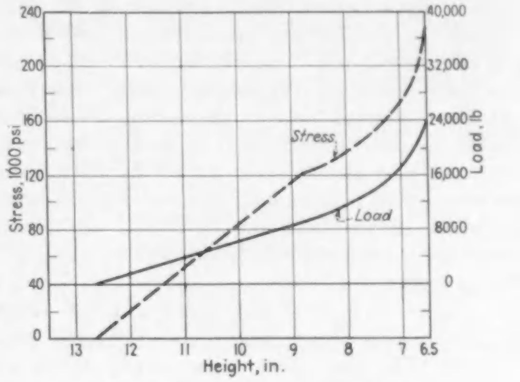
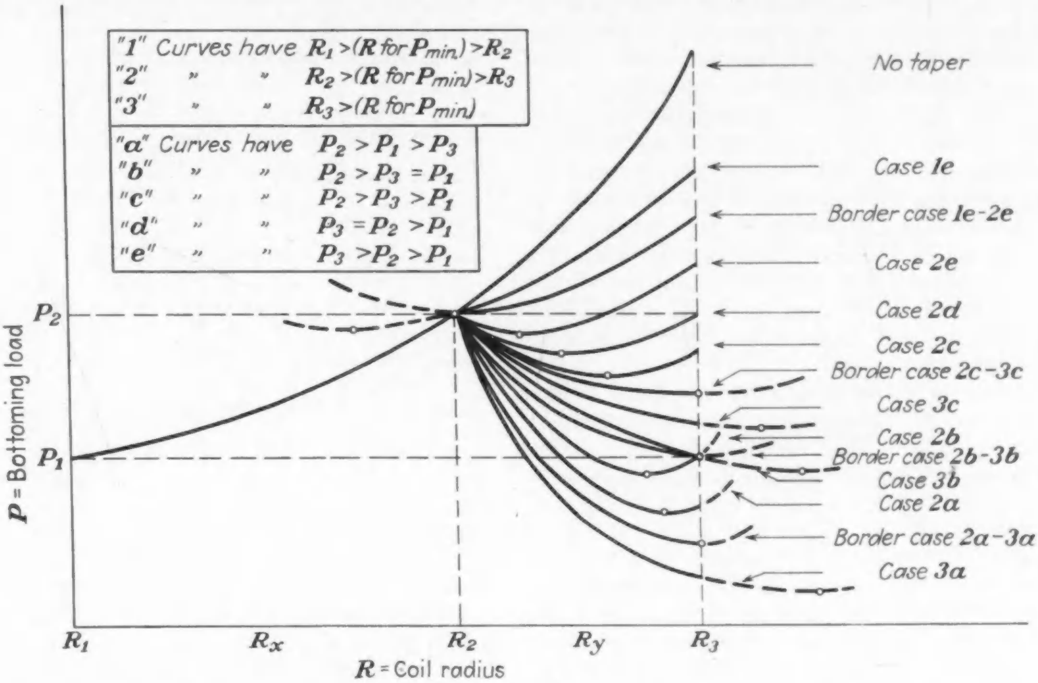


Fig. 21 – Load and stress versus height

■ Figs. 20 and 21 – Curves for a volute spring – 7-in. OD; 6½ x 0.350-in. blade; 4¼ active coils; blade tapered from end of third coil to a thickness of 0.250 in. at inner end

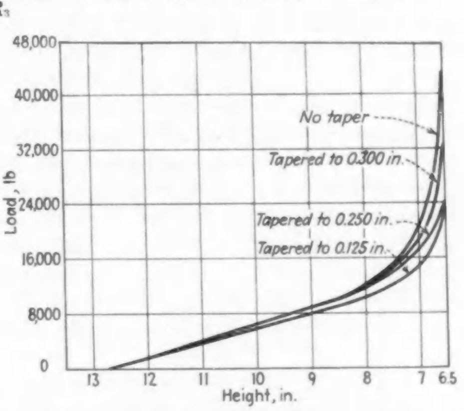


■ Fig. 22 – Schematic diagram showing the bottoming of the coils in various cases of tapering from t_2 to t_3 between R_2 and R_3

until zero thickness is reached at a coil radius R_0. If the tapering is very gradual, the theoretical point for zero thickness may extend beyond the apex of the enveloping paraboloid, and in that event R_0 will be an imaginary value. R_0^2 will then be a negative value which may still be of considerable practical interest for computing the spring.

The points within the active portion of the spring with full blade thickness t_2 continue to be denoted by subscript x, and the points within the active portion of the spring with reduced blade thickness are denoted by subscript y. The basic relationship between L, h, n, R, and t is briefly summarized in the illustration.

The graphs in Figs. 1, 8 and 13 were drawn with the developed coil lengths as abscissa. These coil lengths



■ Fig. 23 – Comparison of load versus spring height curves for volute springs – 7-in. OD; 6½ x 0.350-in. blade; 4¼ active coils; blade tapered from end of third coil to an inner end thickness as shown

involve two variables: n, the number of coils, and R, the coil radii. In the load, stress, and deflection formulas which will be summarized later, the variable n is completely eliminated, and the formulas are built up around the coil radii. Since the coil radii thus become the criterion foi all the important data in the spring, it is a natural conclusion to draw graphs with the decreasing values of coil radii as abscissa. In these graphs each active coil is represented by an equally long piece of abscissa, while in the previous graphs with the developed coil length as abscissa a large outer coil was represented by a longer piece of abscissa than a small inner coil.

Fig. 14 is an example of such a graph for an untapered spring, with two ordinate scales – one for loads and one for stresses. The load curve shows under what load a spring element of a certain coil radius will bottom. The stress curve shows the maximum stress which will occur within a spring element when it bottoms.

In the case of this particular untapered spring the maximum load of 43,700 lb will theoretically be reached when bottoming occurs at the smallest coil radius. However, this maximum load could be attained only with a stress of 291,000 psi which is far beyond the elastic limit.

Fig. 15 shows the theoretical load-deflection curve for this spring, and also a stress curve which indicates the maximum stress at some point in the spring (without designating that point) when any given load is reached.

Fig. 16 shows bottoming loads and stresses against coil radii for a spring of identical dimensions but with the blade thickness tapered from 0.350 in. at the end of the third coil to 0.300 in. at the inner end of the blade. The maximum load has been reduced to 32,400 lb and the maximum stress to 259,000 psi.

Fig. 17 shows the load-deflection curve for this spring which is not materially different in character from that for the untapered spring.

A considerably different picture presents itself in Fig. 18 when the same spring is tapered from 0.350 in. at the end of the third coil to 0.125 in. at the inner end of the blade. No longer does the bottoming load increase from R_1 over R_2 to R_3. The load (P_3) at R_3 has dropped, not only below P_2 but even below P_1. The tapering in this case is so severe that the blade has become too weak at the smallest active coil radius to sustain a load even as much as P_1. The bottoming will now start at R_3 and, after the load has reached the level of P_1, bottoming will progress from the inner as well as from the outer end of the spring. The final bottoming under $P_{max.}$ will occur at R_2. The maximum load in this spring is only 23,800 lb, and the maximum stress is 206,000 psi.

Fig. 19 shows the load-deflection curve. While the maximum load has been considerably reduced from that of the original untapered spring, the full deflection of 6⅛ in. has been maintained in the face of a 30% reduction in the maximum stress. This case differs from that of the actual spring shown in Fig. 12 which ended up with the full deflection reduced by ¾ in. due to settling in the bulldozing operations while the load was not reduced so drastically.

If a volute spring must be designed within the limits of a certain available space, a choice must be made basically to give preference either to a greater maximum load or to a greater total deflection. Sometimes a compromise solution may be the answer. Obviously between the extreme case of very gradual tapering to 0.300 in. (shown in the P-R curve in Fig. 16) and the opposite extreme of a very sharp taper to 0.125 in. (shown in the P-R curve in Fig. 18), there must be intermediate cases of different characteristics. Fig. 22 shows a schematic diagram of the various cases which may arise. All the P-R curves are cubics and have a minimum load value corresponding to some coil radius. The cases 1, 2 and 3 can be distinguished according to the magnitude of this value of (R for $P_{min.}$) relative to the values of R_2 and R_3. Table 2, "Tapering Characteristics" explains this distinction, and also the distinction between classifications a, b, c, d and e.

Table 2 – Tapering Characteristics

In order to understand the general characteristics of any specific tapering case, it is necessary to establish the value of the coil radius for P_{min} in the P - R curve of that part of the spring which is tapered between R_2 and R_3 from t_2 to t_3, and to view this (R for P_{min}) in its relationship to R_2 and R_3:

$$P_y = P_1 \times \frac{R_1^2}{R_y^2} \times \frac{t_y^3}{t_2^3}$$

$$\frac{P_1R_1^2}{t_2^3} \times \frac{\left[t_3R_2^2 - t_2R_3^2 + (t_2 - t_3)\,R_y^2\right]^3}{\left[R_2^2 - R_3^2\right]^3\;R_y^2}$$

Substitute
$$t_3R_2^2 - t_2R_3^2 = A$$
$$t_2 - t_3 = B$$
$$(R_2^2 - R_3^2)^3 = C$$

$$P_y = \text{constant} \times \frac{(A + BR_y^2)^3}{C\,R_y^2}$$

Thus P_y is a function of R_y, and P_y is a minimum when its derivative equals zero:

$$\frac{CR_y^2 \times 3\,(A + BR_y^2)^2 \times 2BR_y - (A + BR_y^2)^3 \times 2CR_y}{C^2R_y^4} = 0$$

$$6BCR_y^3\,(A + BR_y^2)^2 = 2CR_y\,(A + BR_y^2)^3$$

$$3BR_y^2 = A + BR_y^2$$

$$R_y^2 = \frac{A}{2B} = \frac{t_3R_2^2 - t_2R_3^2}{2(t_2 - t_3)}$$

$$P_y = \text{minimum when } (R^2 \text{ for } P_{min}) = \frac{t_3R_2^2 - t_2R_3^2}{2(t_2 - t_3)} = \frac{-R_o^2}{2}$$

In Cases 1:	$(R^2 \text{ for } P_{min}) > R_2^2$	
In Border Cases 1 - 2:	$(R^2 \text{ for } P_{min}) = R_2^2$	
In Cases 2:	$(R^2 \text{ for } P_{min}) < R_2^2$ $> R_3^2$	
In Border Cases 2 - 3:	$(R^2 \text{ for } P_{min}) = R_3^2$	
In Cases 3:	$(R^2 \text{ for } P_{min}) < R_3^2$	
In Cases "a" :	$P_3 = P_1 \times \frac{R_1^2}{R_3^2} \times \frac{t_3^3}{t_2^3} < P_1$	$\frac{t_2^3}{t_3^3} R_3^2 > R_1^2$
In Cases "b" :	$P_3 = P_1 \times \frac{R_1^2}{R_3^2} \times \frac{t_3^3}{t_2^3} = P_1$	$\frac{t_2^3}{t_3^3} R_3^2 = R_1^2$
In Cases "c" :	$P_3 = P_1 \times \frac{R_1^2}{R_3^2} \times \frac{t_3^3}{t_2^3} = P_x = P_1 \frac{R_1^2}{R_x^2}$	$\frac{t_2^3}{t_3^3} R_3^2 = R_x^2$
In Cases "d" :	$P_3 = P_1 \times \frac{R_1^2}{R_3^2} \times \frac{t_3^3}{t_2^3} = P_2 = P_1 \frac{R_1^2}{R_2^2}$	$\frac{t_2^3}{t_3^3} R_3^2 = R_2^2$
In Cases "e" :	$P_3 = P_1 \times \frac{R_1^2}{R_3^2} \times \frac{t_3^3}{t_2^3} = P_y > P_1 \frac{R_1^2}{R_2^2}$	$\frac{t_2^3}{t_3^3} R_3^2 < R_2^2$

When the stresses and the load-deflection curve for a tapered spring of given dimensions are to be computed, the tapering case must be identified before it is possible to pick out the proper set of formulas, depending on whether the bottoming of the tapered coils will start at R_2 (cases "1") or at R_3 (cases "3").

The cases "2" introduce an additional complication because there the bottoming of the tapered coils starts neither at R_2 nor at R_3 but at some point between the two. In the cases 2a, 2b, and 2c the bottoming will spread from this point towards R_3 and towards a coil radius R_{v_3} where bottoming will occur simultaneously with the bottoming at R_3. The entire portion between R_{v_3} and R_3 then becomes inactive, and the remaining active portion of the spring between R_1 and R_{v_3} may thereafter be treated like a case "3."

In the case 2e the bottoming will spread both toward R_2 and towards a coil radius R_{y_2} where bottoming will occur simultaneously with the bottoming at R_2. The entire portion between R_2 and R_{y_2} then becomes inactive, and since the untapered portion between R_1 and R_2 has also become inactive, the only portion which remains active is that between R_{y_2} and R_3, and this may thereafter be treated like a case "1."

A case "2c" is presented in Fig. 20 where the same spring which was first used without tapering in Fig. 14 is now tapered from 0.350 in. at the end of the third coil to 0.250 in. at the inner end of the blade. Fig. 21 shows the load-deflection curve for this spring.

In Fig. 23 the load-deflection curve for the untapered spring and the three curves for the tapering cases "1e" (to 0.300 in.), "2c" (to 0.250 in.), and "3a" (to 0.125 in.) are combined to show the extent of the discrepancy between the curves. The graphs furnish ample explanations for the tapering cases.

This chapter will be concluded with a series of formulas from which untapered and partly tapered springs may be computed. The formulas for loads required to cause bottoming at various points in the spring are presented in Fig. 24, and the formulas for maximum stresses corresponding to these loads are given in Fig. 25.

The derivation, Table 3 (see following page), explains the formulas for the tapered part of a "case 1e" volute spring, and Fig. 26 gives the formulas for deflections both in an untapered spring and in a "case 1e" spring.

The derivation, Table 4 (see p. 237), explains the formulas for the tapered part of a "case 3a" volute spring, and Fig. 27 gives the formulas for deflections in this spring.

The Appendix[1] to the paper contains further sets of formulas for deflections in springs of tapering cases 3b, 3c, 2a, 2c, 2e. The remaining cases may be easily derived from these sets of formulas.

The Appendix also gives derivations for the following two weight formulas:

$$\text{Weight of an untapered spring} = 0.283\,(R_1 + R_2)\,\pi b t\left(n + \frac{15}{16}\right)$$

Weight of a partly tapered spring =

$$0.283\pi b\left[(R_1 + R_2)n_2 t_2 + (R_2 + R_3)n_3\,\frac{t_2 + t_3}{2} + \frac{15}{16}\,R_1 t_2 + \frac{3}{2}\,R_3 t_3 - \frac{9}{8}\,R_3{}^2\,\frac{t_2 - t_3}{n_3(R_2 + R_3)}\right]$$

The subject of tapering the blade thickness near the inner end of the active coils has been treated here in some detail because it appears to offer the best opportunity for changing the characteristics of a spring of given size to a considerable extent. It is, however, well to bear in mind that in accordance with the strain energy theory a certain quantity of steel under a certain maximum stress can store only so much and no more energy. In the case of the volute spring, the picture is complex because the stresses are unequal. However, there are always some portions of the spring which will be subjected to the maximum stress and, at least for that portion, the maximum amount of energy which can be stored in the material imposes limitations either on the maximum deflection or on the maximum load for the entire spring even under the best available stress conditions.

The requirements for maximum load are not always known or realized; yet their importance is of the first order in the design of any spring and, more particularly, in the design of the volute spring which is given preference over other spring types in many cases because it is presumed to have higher load-carrying capacity due to its increasing rate.

There are, of course, other important aspects in the design and in the manufacture of volute springs, such as the combined question of loading eccentricity and friction, on which we hope to gain more valuable information as our laboratory work progresses. At present, our laboratory work on the volute spring is not yet sufficiently comprehensive to serve as a basis for comparisons with results in actual service; we therefore feel that information on field operations would not serve its proper purpose and should be withheld. We are, however, doing our best to establish a full complement of laboratory information.

Most prominent in this work is the life testing of the springs. To one who has conducted spring life tests and has been guided by their results for many years it is difficult to see eye to eye with those engineers who profess their disbelief in the value of these tests. Yet it might as well be admitted that the spring life tests have widely fallen into disrepute because they are rarely conducted in a scientifically acceptable manner. And conclusions drawn from wrong test premises are dangerous half-truths.

It is generally agreed that life tests on any material specimen and on any machine element should be "constant-stress" tests. All scientific investigations on endurance limits and on stress-cycle curves showing the approach to the endurance limit presuppose an exactly controlled stress condition in the test piece. Any work which is to fit into the sum total of past experience and which is to find

LOAD REQUIRED TO "BOTTOM" POINT OF SUBSCRIPT	VOLUTE SPRING WITH t_2 CONSTANT BETWEEN R_1 & R_3	VOLUTE SPRING WITH t_2 CONSTANT BETWEEN R_1 & R_2. TAPERED TO t_3 BETWEEN R_2 & R_3
P (NO BOTTOMING)	$\frac{P}{P_1}\,\frac{\eta_3 b t_2^3 G \tan\alpha}{R_1^2} = P_1 \times \frac{P}{P_1}$	(SAME)
P_1	$\frac{\eta_3 b t_2^3 G \tan\alpha}{R_1^2}$	(SAME)
P_x	$\frac{\eta_3 b t_2^3 G \tan\alpha}{R_x^2} = P_1 \times \frac{R_1^2}{R_x^2}$	(SAME)
P_2	$\frac{\eta_3 b t_2^3 G \tan\alpha}{R_2^2} = P_1 \times \frac{R_1^2}{R_2^2}$	(SAME)
P_y	$\frac{\eta_3 b t_2^3 G \tan\alpha}{R_y^2} = P_1 \times \frac{R_1^2}{R_y^2}$	$\frac{\eta_3 b t_y^3 G \tan\alpha}{R_y^2} = P_1 \times \frac{R_1^2}{R_y^2} \times \frac{t_y^3}{t_2^3}$
P_3	$\frac{\eta_3 b t_2^3 G \tan\alpha}{R_3^2} = P_1 \times \frac{R_1^2}{R_3^2}$	$\frac{\eta_3 b t_3^3 G \tan\alpha}{R_3^2} = P_1 \times \frac{R_1^2}{R_3^2} \times \frac{t_3^3}{t_2^3}$

■ Fig. 24 – Formulas for loads required to "bottom" certain points in the volute spring

MAXIMUM TORSIONAL STRESS WHEN POINT OF SUBSCRIPT IS "BOTTOMING"	VOLUTE SPRING WITH t_2 CONSTANT BETWEEN R_1 & R_3	VOLUTE SPRING WITH t_2 CONSTANT BETWEEN R_1 & R_2 TAPERED TO t_3 BETWEEN R_2 & R_3
τ_c (NO BOTTOMING)	$\frac{P_2 R_1}{\eta_2 b t_2^2} K_1 = \frac{P}{P_1} \times \tau_{c_1}$	(SAME)
τ_{c_1}	$\frac{P_1 R_1}{\eta_2 b t_2^2} K_1 = \frac{\eta_3}{\eta_2}\,\frac{G \tan\alpha\, t_2}{R_1} K_1$	(SAME)
τ_{c_x}	$\frac{P_x R_x}{\eta_2 b t_2^2} K_x = \frac{\eta_3}{\eta_2}\,\frac{G \tan\alpha\, t_2}{R_x} K_x$	(SAME)
τ_{c_2}	$\frac{P_2 R_2}{\eta_2 b t_2^2} K_2 = \frac{\eta_3}{\eta_2}\,\frac{G \tan\alpha\, t_2}{R_2} K_2$	(SAME)
τ_{c_y}	$\frac{P_y R_y}{\eta_2 b t_2^2} K_y = \frac{\eta_3}{\eta_2}\,\frac{G \tan\alpha\, t_2}{R_y} K_y$	$\frac{P_y R_y}{\eta_2 b t_y^2} K'_y = \frac{\eta_3}{\eta_2}\,\frac{G \tan\alpha\, t_y}{R_y} K'_y$
τ_{c_3}	$\frac{P_3 R_3}{\eta_2 b t_2^2} K_3 = \frac{\eta_3}{\eta_2}\,\frac{G \tan\alpha\, t_2}{R_3} K_3$	$\frac{P_3 R_3}{\eta_2 b t_3^2} K'_3 = \frac{\eta_3}{\eta_2}\,\frac{G \tan\alpha\, t_3}{R_3} K'_3$

■ Fig. 25 – Formulas for maximum torsional stresses corresponding to "bottoming" loads

Table 3 – Derivation of Formulas for the Part of Volute Spring Tapered from t_2 to t_3 between R_2 and R_3 (Case 1e)
Symbols correspond to those of Fig. 13.

1. GEOMETRY

(η_2 and η_3 considered constant)

$$L_y = (R_2+R_y)\pi n_y = \frac{\pi n_3(R_2^2-R_y^2)}{R_2-R_3} \qquad \frac{dL_y}{dR_y} = \frac{-2\pi n_3 R_y}{R_2-R_3}$$

$$h_y = \frac{h_3 L_y}{L_3} = \frac{h_3(R_2^2-R_y^2)}{R_2^2-R_3^2} \qquad \frac{dh_y}{dR_y} = \frac{-2h_3 R_y}{R_2^2-R_3^2}$$

$$\frac{dh_y}{dL_y} = \frac{dh_y}{dR_y} \times \frac{dR_y}{dL_y} = \frac{h_3}{\pi n_3(R_2+R_3)}$$

2. LOADS

$$\frac{df}{dL_y} = \frac{R_y\, d\phi}{dL_y} = \frac{P_y R_y^2}{G\eta_3 b t_y^3}$$

when R_y is bottomed, $\frac{df}{dL_y} = \frac{dh_y}{dL_y}$

$$\frac{P_y R_y^2}{G\eta_3 b t_y^3} = \frac{h_3}{\pi n_3(R_2+R_3)} = \tan x$$

$$P_y = \frac{G\eta_3 b t_y^3 \tan\alpha}{R_y^2}$$

$$P_2 = P_{tr} = \frac{G\eta_3 b t_2^3 \tan\alpha}{R_2^2}$$

$$P_3 = P_{max} = \frac{G\eta_3 b t_3^3 \tan\alpha}{R_3^2}$$

3. STRESSES

$$T_{c_y} = \frac{P_y R_y}{\eta_2 b t_y^2} K_y = \frac{\eta_3}{\eta_2}\,\frac{G \tan\alpha\, t_y}{R_y} K_y$$

4. DEFLECTIONS

a) Load $P < P_{tr}$

$$df = dL_y \frac{df}{dL_y} = \frac{PR_y^2}{G\eta_3 b t_y^3}\left[\frac{-2\pi n_3 R_y}{R_2-R_3}(dR_y)\right]$$

$$= \frac{-2\pi n_3 P(R_2^2-R_3^2)^3 R_y^3 (dR_y)}{G\eta_3 b(R_2-R_3)\left[t_3R_2^2-t_2R_3^2+(t_2-t_3)R_y^2\right]^3}$$

$$f = \frac{P\pi n_3(R_2^2-R_3^2)^3}{G\eta_3 b(R_2-R_3)}\int_{R_y=R_2}^{R_y=R_3} - \frac{2R_y^3(dR_y)}{\left[t_3R_2^2-t_2R_3^2+(t_2-t_3)R_y^2\right]^3}$$

Substitute: $\left[t_3R_2^2-t_2R_3^2\right] = A$

$(t_2-t_3) = B$

$R_y^2 = Z$

$d(R_y^2) = 2R_y(dR_y) = dZ$

$$f = \frac{P\pi n_3(R_2^2-R_3^2)^3}{G\eta_3 b(R_2-R_3)} \times -\int_{R_2}^{R_3}\frac{Z\,dZ}{(A+BZ)^3}$$

$$f = \frac{P\pi n_3(R_2^2-R_3^2)^3}{G\eta_3 b(R_2-R_3)} \times \frac{1}{2B^2}\left[\frac{A+2BR_3^2}{(A+BR_3^2)^2} - \frac{A+2BR_2^2}{(A+BR_2^2)^2}\right]$$

$$f = \frac{P\pi n_3(R_2^2-R_3^2)^3}{G\eta_3 b(R_2-R_3)} \times \frac{(t_3R_2^2+t_2R_3^2)}{2t_2^2t_3^2(R_2^2-R_3^2)^2}$$

$$f_{tr} = P_2 \times \frac{\pi n_3(R_2^2-R_3^2)^3}{G\eta_3 b(R_2-R_3)} \times \frac{(t_3R_2^2+t_2R_3^2)}{2t_2^2t_3^2(R_2^2-R_3^2)^2}$$

$$f_{tr} = h_3\,\frac{\frac{t_2}{t_3}R_2^2+\frac{t_2^2}{t_3^2}R_3^2}{2R_2^2} = \frac{P_2}{P_1}\times h_3\,\frac{\frac{t_2}{t_3}R_2^2+\frac{t_2^2}{t_3^2}R_3^2}{2R_1^2}$$

b) Load $P_y \begin{smallmatrix} > P_{tr} \\ < P_{max}\end{smallmatrix}$

$$f_y' = h_3 \times \frac{n_y}{n_3} \times \frac{R_2+R_y}{R_2+R_3} = h_3\,\frac{R_2^2-R_y^2}{R_2^2-R_3^2}$$

$$f_y'' = P_y \times \frac{\pi(n_3-n_y)(R_y^2-R_3^2)^3}{G\eta_3 b(R_y-R_3)} \times \frac{(t_3R_y^2+t_yR_3^2)(R_y+R_3)}{2t_y^2t_3^2(R_y^2-R_3^2)^2}$$

$$f_y'' = h_3\,\frac{\left(\frac{t_y}{t_3}R_y^2+\frac{t_y^2}{t_3^2}R_3^2\right)(R_y^2-R_3^2)}{2R_y^2(R_2^2-R_3^2)}$$

$$f_y = f_y' + f_y''$$

Table 4 – Derivation of Formulas for the Part of Volute Spring Tapered from t_2 to t_3 between R_2 and R_3 (Case 3a)
Symbols correspond to those of Fig. 13.

1. GEOMETRY (η_2 and η_3 considered constant)

$$L_3-L_y = (R_y+R_3)\pi(n_3-n_y) = \frac{\pi n_3(R_y^2-R_3^2)}{R_2-R_3} \qquad \frac{d(L_3-L_y)}{dR_y} = \frac{2\pi n_3 R_y}{R_2-R_3}$$

$$h_3-h_y = \frac{h_3(L_3-L_y)}{L_3} = \frac{h_3(R_y^2-R_3^2)}{R_2^2-R_3^2} \qquad \frac{d(h_3-h_y)}{dR_y} = \frac{2h_3R_y}{R_2^2-R_3^2}$$

$$\frac{d(h_3-h_y)}{d(L_3-L_y)} = \frac{d(h_3-h_y)}{dR_y}\times\frac{dR_y}{d(L_3-L_y)} = \frac{h_3}{\pi n_3(R_2+R_3)}$$

2. LOADS

$$\frac{df}{d(L_3-L_y)} = \frac{R_y\,d\phi}{d(L_3-L_y)} = \frac{P_yR_y^2}{G\eta_3 b t_y^3}$$

when R_y is bottomed, $\dfrac{df}{d(L_3-L_y)} = \dfrac{d(h_3-h_y)}{d(L_3-L_y)}$

$$\frac{P_yR_y^2}{G\eta_3 b t_y^3} = \frac{h_3}{\pi n_3(R_2+R_3)} = \tan\alpha$$

$$P_y = \frac{G\eta_3 b t_y^3\tan\alpha}{R_y^2}$$

$$P_3 = P_{tr} = \frac{G\eta_3 b t_3^3\tan\alpha}{R_3^2}$$

$$P_2 = P_{max.} = \frac{G\eta_3 b t_2^3\tan\alpha}{R_2^2}$$

3. STRESSES

$$T_{c_y} = \frac{P_yR_y}{\eta_2 b t_y^2}K_y = \frac{\eta_3}{\eta_2}\,\frac{G\tan\alpha\, t_y}{R_y}K_y$$

4. DEFLECTIONS

a) Load $P<P_{tr}$

$$df = d(L_3-L_y)\frac{df}{d(L_3-L_y)} = \frac{PR_y^2}{G\eta_3 b t_y^3}\left[\frac{2\pi n_3 R_y}{R_2-R_3}(dR_y)\right]$$
$$= \frac{2\pi n_3 P(R_2^2-R_3^2)^3R_y^3(dR_y)}{G\eta_3 b(R_2-R_3)\left[t_3R_2^2-t_2R_3^2+(t_2-t_3)R_y^2\right]^3}$$

$$f = \frac{P\pi n_3(R_2^2-R_3^2)^3}{G\eta_3 b(R_2-R_3)}\int_{R_y=R_3}^{R_y=R_2}\frac{2R_y^3(dR_y)}{\left[t_3R_2^2-t_2R_3^2+(t_2-t_3)R_y^2\right]^3}$$

substitute: $[t_3R_2^2-t_2R_3^2]=A \qquad (t_2-t_3)=B$

$R_y^2=Z \qquad d(R_y^2)=2R_y(dR_y)=dZ$

$$f = \frac{P\pi n_3(R_2^2-R_3^2)^3}{G\eta_3 b(R_2-R_3)}\int_{R_3}^{R_2}\frac{Z\,dZ}{(A+BZ)^3}$$

$$f = \frac{P\pi n_3(R_2^2-R_3^2)^3}{G\eta_3 b(R_2-R_3)}\times\frac{1}{2B^2}\left[\frac{A+2BR_3^2}{(A+BR_3^2)^2}-\frac{A+2BR_2^2}{(A+BR_2^2)^2}\right]$$

$$f = \frac{P\pi n_3(R_2^2-R_3^2)^3}{G\eta_3 b(R_2-R_3)}\times\frac{(t_3R_2^2+t_2R_3^2)}{2t_2^2t_3^2(R_2^2-R_3^2)^2}$$

$$f = P\times\frac{\pi n_3}{2G\eta_3 b}\times\frac{t_3R_2^2+t_2R_3^2}{t_2^2\,t_3^2}\times(R_2+R_3)$$

$$f_3 = f_{tr} = P_3\times\frac{\pi n_3}{2G\eta_3 b}\times\frac{t_3R_2^2+t_2R_3^2}{t_2^2\,t_3^2}\times(R_2+R_3)$$

$$f_{tr} = \frac{h_3G\eta_3 b t_3^3}{\pi n_3R_3^2(R_2+R_3)}\times\frac{\pi n_3(R_2+R_3)}{2G\eta_3 b}\times\frac{t_3R_2^2+t_2R_3^2}{t_2^2\,t_3^2}$$

$$f_{tr} = h_3\frac{\frac{t_3}{t_2}R_3^2+\frac{t_3^2}{t_2^2}R_2^2}{2R_3^2} = \frac{P_3}{P_1}\times h_3\frac{\frac{t_3^2}{t_3^2}R_3^2+\frac{t_3}{t_1}R_2^2}{2R_1^2}$$

b) Load $P_3 \gtrless {P_{tr} \atop P_{max.}}$

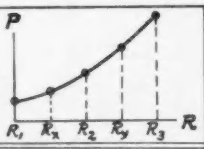
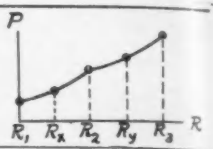
$$f_y' = h_3\times\frac{n_y}{n_3}\times\frac{R_y+R_3}{R_2+R_3} = h_3\frac{R_y^2-R_3^2}{R_2^2-R_3^2}$$

$$f_y'' = P_y\times\frac{\pi n_y}{2G\eta_3 b}\times\frac{t_yR_2^2+t_2R_y^2}{t_2^2\,t_y^2}\times(R_2+R_y)$$

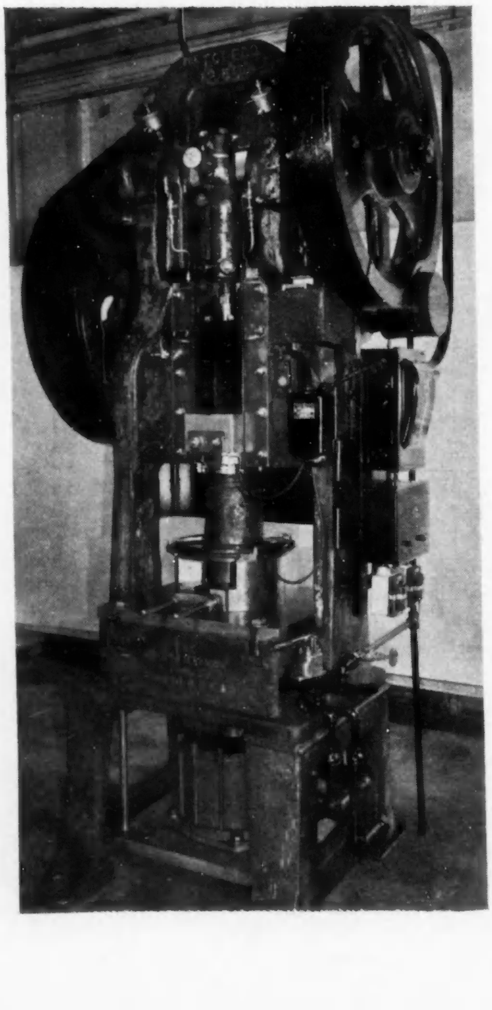
$$f_y'' = \frac{t_y^3R_2^2}{t_2^3R_y^2}\times\frac{h_3\eta_3Gbt_2^3}{\pi n_3R_2^2(R_2+R_3)}\times\frac{\pi(R_2+R_3)}{2G\eta_3 b}\times\frac{n_3(R_2-R_y)}{R_2-R_3}\times\frac{t_yR_2^2+t_2R_y^2}{t_2^2t_y^2}$$

$$f_y'' = h_3\frac{\left(\frac{t_y}{t_2}R_y^2+\frac{t_y^2}{t_2^2}R_2^2\right)(R_2^2-R_y^2)}{2R_y^2(R_2^2-R_3^2)}$$

$$f_y = f_y'+f_y''$$

DEFLECTION WHEN POINT OF SUBSCRIPT IS "BOTTOMING"	VOLUTE SPRING WITH t_2 CONSTANT BETWEEN R_1 AND R_3		VOLUTE SPRING WITH t_2 CONSTANT BETWEEN R_1 AND R_2, TAPERED TO t_3 BETWEEN R_2 AND R_3. CASE Ie: $(R \text{ for } P_{min}) > R_2$; $P_3 > P_2 > P_1$			
			COILS BETWEEN R_1 & R_2		COILS BETWEEN R_2 & R_3	
	BOTTOMED COILS	FREE COILS	BOTTOMED COILS	FREE COILS	BOTTOMED COILS	FREE COILS
	f'_x	f''_x	f'_x	f''_x	f'_y	f''_y
f (NO BOTTOMING)		$\frac{P}{P_1}\left[(h_2+h_3)\cdot\frac{R_1^2+R_3^2}{2R_1^2}\right]$		$\frac{P}{P_1}\left[h_2\frac{R_1^2+R_2^2}{2R_1^2}\right]$		$+\frac{P}{P_1}\left[h_3\frac{\frac{t_2}{t_3}R_2^2+\frac{t_2^2}{t_3^2}R_3^2}{2R_1^2}\right]$
f_1		$(h_2+h_3)\frac{R_1^2+R_3^2}{2R_1^2}$		$h_2\frac{R_1^2+R_2^2}{2R_1^2}$		$+\ h_3\frac{\frac{t_2}{t_3}R_2^2+\frac{t_2^2}{t_3^2}R_3^2}{2R_1^2}$
f_x	$(h_2+h_3)\frac{R_1^2-R_x^2}{R_1^2-R_3^2}$	$+(h_2+h_3)\frac{(R_x^2+R_3^2)(R_x^2-R_3^2)}{2R_x^2(R_1^2-R_3^2)}$	$h_2\frac{R_1^2-R_x^2}{R_1^2-R_2^2}$	$+h_2\frac{(R_x^2+R_2^2)(R_x^2-R_2^2)}{2R_x^2(R_1^2-R_2^2)}$		$+\ h_3\frac{\frac{t_2}{t_3}R_2^2+\frac{t_2^2}{t_3^2}R_3^2}{2R_x^2}$
f_2	$(h_2+h_3)\frac{R_1^2-R_2^2}{R_1^2-R_3^2}$	$+(h_2+h_3)\frac{(R_2^2+R_3^2)(R_2^2-R_3^2)}{2R_2^2(R_1^2-R_3^2)}$	h_2			$+\ h_3\frac{\frac{t_2}{t_3}R_2^2+\frac{t_2^2}{t_3^2}R_3^2}{2R_2^2}$
f_y	$(h_2+h_3)\frac{R_1^2-R_y^2}{R_1^2-R_3^2}$	$+(h_2+h_3)\frac{(R_y^2+R_3^2)(R_y^2-R_3^2)}{2R_y^2(R_1^2-R_3^2)}$	h_2		$+h_3\frac{R_2^2-R_y^2}{R_2^2-R_3^2}$	$+h_3\frac{\left(\frac{t_y}{t_3}R_y^2+\frac{t_y^2}{t_3^2}R_3^2\right)(R_y^2-R_3^2)}{2R_y^2(R_2^2-R_3^2)}$
f_3	h_2+h_3		h_2		$+\ h_3$	

■ Fig. 26 (above) – Deflection formulas for untapered spring and for tapering case Ie



■ Fig. 28 (left) – Toledo press converted into spring test machine

consideration in future studies will have to take this prerequisite into account.

This condition is fairly easy to meet when the test cycle extends over the full range of equal stresses with opposite signs where the mean stress is at zero. In a compression spring such a cycle cannot be used. Since the test should bear a semblance to the actual operating conditions, the spring must be tested between a minimum and a maximum compression load in such a way that the static load point falls near the half-way mark in the amplitude of the cycle, and that the maximum load and the maximum stress conform with the values reached in actual service, unless a test result is sought for some other specific value.

Such a cycle can be computed and set up on the test machine for any spring. When it is set up on a "constant-stroke" machine such as a press, it will be correct only as long as the spring does not take a permanent set. When the spring begins to settle the loads and stresses are reduced, and the base line for the test is lost.

Our own laboratory work with suspension springs is conducted on a "drop-weight" machine which Tore Franzen developed many years ago. A static weight connected with the spring is lifted a certain distance by means of a cam and then drops freely and oscillates the spring in its natural frequency. The static load and stress remains constant, and a frequent check and correction of the lift of the weight

VOLUTE SPRING WITH t_2 CONSTANT BETWEEN R_1 & R_2, TAPERED TO t_3 BETWEEN R_2 & R_3

CASE 3a: $R_3 > (R \text{ for } P_{MIN})$; $P_2 > P_1 > P_3$

DEFLECTION WHEN POINT OF SUBSCRIPT IS "BOTTOMING"	COILS BETWEEN R_1 & R_2		COILS BETWEEN R_2 & R_3	
	BOTTOMED COILS	FREE COILS	FREE COILS	BOTTOMED COILS
	f'_x	f''_x	f''_y	f'_y
f (NO BOTTOMING)		$\frac{P}{P_1}\left[h_2 \frac{R_1^2+R_2^2}{2R_1^2}\right]$	$+\frac{P}{P_3}\left[h_3 \frac{\frac{t_3^2}{t_2^2}R_2^2+\frac{t_3}{t_2}R_3^2}{2R_3^2}\right]$	
f_3		$\frac{P_3}{P_1}\left[h_2 \frac{R_1^2+R_2^2}{2R_1^2}\right]$	$+ h_3 \frac{\frac{t_3^2}{t_2^2}R_2^2+\frac{t_3}{t_2}R_3^2}{2R_3^2}$	
f_y for $P_y {> P_3 \atop < P_1}$		$\frac{P_y}{P_1}\left[h_2 \frac{R_1^2+R_2^2}{2R_1^2}\right]$	$+ h_3 \frac{\left(\frac{t_y}{t_2}R_y^2+\frac{t_y^2}{t_2^2}R_2^2\right)(R_2^2-R_y^2)}{2R_y^2(R_2^2-R_3^2)}$	$+h_3 \frac{R_y^2-R_3^2}{R_2^2-R_3^2}$
f_1		$h_2 \frac{R_1^2+R_2^2}{2R_1^2}$	$+ h_3 \frac{\left(\frac{t_{y(1)}}{t_2}R_{y(1)}^2+\frac{t_{y(1)}^2}{t_2^2}R_2^2\right)(R_2^2-R_{y(1)}^2)}{2R_{y(1)}^2(R_2^2-R_3^2)}$	$+h_3 \frac{R_{y(1)}^2-R_3^2}{R_2^2-R_3^2}$
f_x for $P_x {> P_3 \atop {> P_1 \atop < P_2}}$	$h_2 \frac{R_1^2-R_x^2}{R_1^2-R_2^2}$	$+h_2 \frac{(R_x^2+R_2^2)(R_x^2-R_2^2)}{2R_x^2(R_1^2-R_2^2)}$	$+ h_3 \frac{\left(\frac{t_{y(x)}}{t_2}R_{y(x)}^2+\frac{t_{y(x)}^2}{t_2^2}R_2^2\right)(R_2^2-R_{y(x)}^2)}{2R_{y(x)}^2(R_2^2-R_3^2)}$	$+h_3 \frac{R_{y(x)}^2-R_3^2}{R_2^2-R_3^2}$
f_2 for $P_{MAX}=P_2$	h_2			$+h_3$

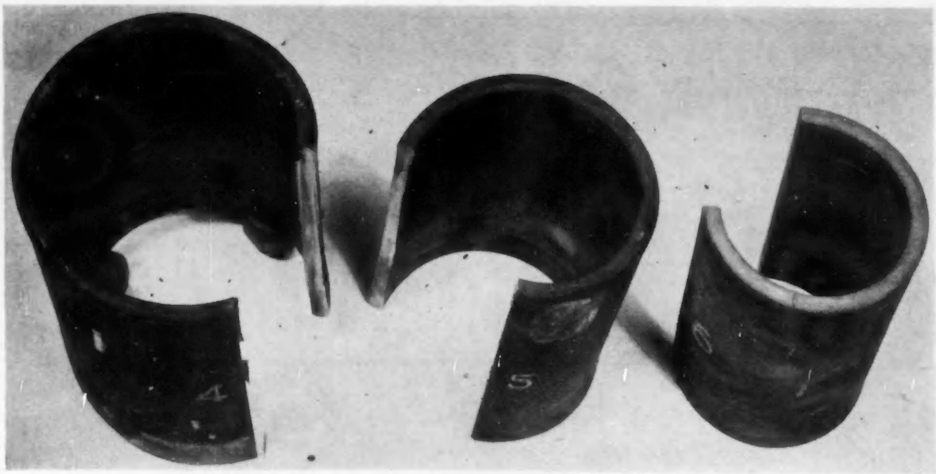
■ Fig. 27 (above) – Deflection formulas for tapering case 3a

assure a reasonable constancy of the minimum and maximum stress values. However, the free oscillation of the spring adds a number of smaller cycles to each main cycle. They are difficult to control and often have some influence upon the test result.

While we consider this machine superior to the ordinary constant-stroke press, its capacity is limited, and the large volute springs which required test loads upward of 20,000 lb could not be set up on it. In order to avoid delay and to get a start with some life test, even of a less-desirable type, an old Toledo press was converted into a test machine a year ago (Fig. 28). A new crankshaft was built to give a stroke in accordance with the specifications to which we were working. A movable platform was introduced which is first lifted by means of an air cylinder and a piston and then set down on a substantial pair of blocks, thus preloading the spring. Starting and stopping are effected with the clutch between the flywheel and crankshaft disengaged. The clutch engagement is controlled by a solenoid upon which several safety switches can act. The machine is makeshift but it has made the desired impression upon the springs. Fig. 29 shows a typical volute spring failure which resulted from this life-test work.

Fig. 30 shows parts of a spring which did not completely break through but on which at least one serious crack appeared in the course of the life test. For closer study of the failure, the spring was cut into 7 parts, and each part was investigated by the magnaflux method. It was then found that a number of cracks had started between the smallest active coil marked "6" and the next larger coils marked "5" and "4." The originally visible crack on

■ Fig. 29 – Typical spring failure resulting from life-test work



■ Fig. 30 – Fractures in three spring coils

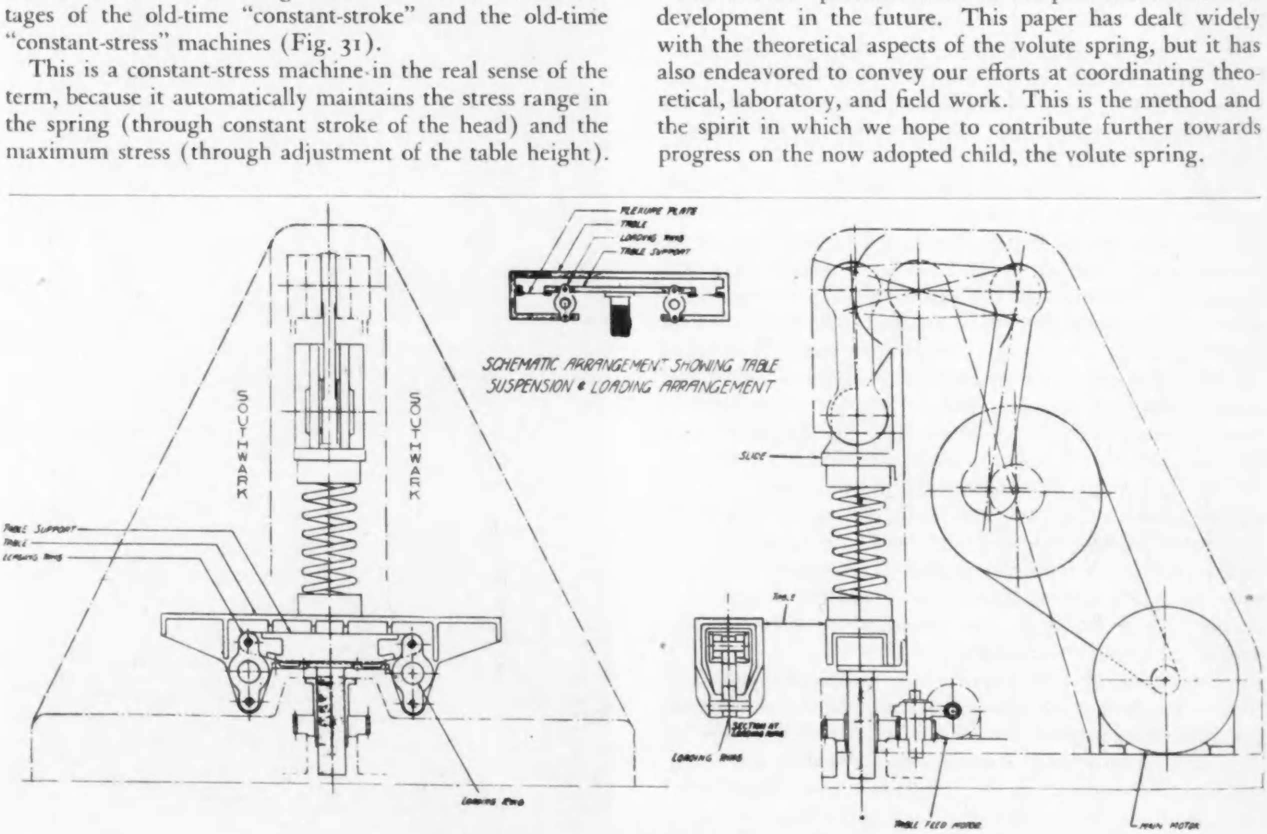
the edge of coil "6" is unusual because it has started at a point which is distant from the theoretical maximum stress region at the middle of the long surface. This result may have been due to a material defect or to a local stress peak brought about by friction or by some other factor. The other cracks follow the usual pattern more closely.

These are just a few samples of the ground work that has been laid for a systematic study of the volute spring. A more adequate machine is, of course, needed for the task. For years we have fully realized the shortcomings of previously available machines and have endeavored to obtain something better. Now we have collaborated with the Baldwin Southwark Division of the Baldwin Locomotive Works in a new design which combines the advantages of the old-time "constant-stroke" and the old-time "constant-stress" machines (Fig. 31).

This is a constant-stress machine in the real sense of the term, because it automatically maintains the stress range in the spring (through constant stroke of the head) and the maximum stress (through adjustment of the table height).

The stroke of the head is set before the test and thereafter remains undisturbed. The table can be lifted by means of a screw and is suspended with two calibrated loading rings which are stressed in tension under the vertical test load. The electrical controls of the machine provide maintenance of the original table setting until spring settling occurs. A light beam travels parallel with the deflection of the loading rings and is thus a measuring device for the loads and the stresses in the spring at every instance of the test. An electric light cell is set in a position where the light beam will just strike it when the predetermined maximum load is reached. Failure of the light beam to strike the cell is an indication that the maximum load has not been reached and that therefore the spring must have settled or failed. When the cell remains dark, it acts upon the table feed motor which lifts the table and compresses the spring, thus restoring the load until the light beam again makes contact with the cell. The entire process is automatic so that actual maintenance of the maximum stress is vouchsafed.

The machine is expected to be in operation by early spring and to give us an opportunity for a check on the theoretical and practical work of the past and for further development in the future. This paper has dealt widely with the theoretical aspects of the volute spring, but it has also endeavored to convey our efforts at coordinating theoretical, laboratory, and field work. This is the method and the spirit in which we hope to contribute further towards progress on the now adopted child, the volute spring.



■ Fig. 31 – Schematic sketch of new life-test machine

GROUND Versus FLIGHT TESTS of Airplane Engine Installations

by JAMES B. KENDRICK
Vega Aircraft Corp.

THE disadvantages of present methods of proving engine installations by flight tests are discussed in this paper. Some data are given to show the great expense of such methods. The conclusion is reached that adequate ground-test facilities should be provided for use in pre-flight development and service tests of new engine installations.

A comparison of the results of ground tests on the Vega Ventura engine installation with flight-test results indicates some factors in ground-test technique which should be satisfied in order to insure reliable results. Similitude conditions to be met for cooling, vibration, and accelerated service tests are discussed to illustrate the method of approach for such problems. Various types of test equipment are described for attaining these conditions, the closed-return wind tunnel appearing to offer the greatest advantages for general testing. A new compact arrangement for a closed-return wind tunnel is described, which will reduce the cost of construction appreciably.

Arguments are presented in favor of the engine test wind tunnel for thorough pre-flight proving of new installations. Further data are given to show the justification for such wind tunnels, due to reductions in the cost of flight testing, as well as avoidance of delays in production, lost sales, and service replacements in the field.

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REASONS for Pre-Flight Testing – There are several reasons for perfecting integral units of the airplane, such as the engine installation, by means of ground tests before they are flown on the airplane. The most important reason is to save flight-test time, permitting more attention to be paid to strictly airplane problems during the flight program and thereby expediting approval of the design project for production. Items such as performance determination, stability, stall characteristics, control forces, airplane vibration, soundproofing, and ventilation, which can finally be verified only in flight, make the program extensive without the added complication of detail development problems on the engine installation and other units of the airplane.

The average length of time required to develop an engine installation by flight testing is greater than that required to determine and perfect all other characteristics of the airplane. The value of this flight time, as shown in Fig. 1, varies greatly with the size of the machine, amounting to several thousand dollars per hour for a large, multi-engined airplane, when personnel, equipment, fuel, changes, insurance, and so on, are considered. However, *these "direct costs" of flight testing are insignificant, compared with the potential losses due to delays in production caused by unforeseen developments of the flight-test program. See Fig. 2.* The value of the weeks or months of elapsed time spent in testing the engine installation will usually be far greater to the company and the customer than the actual cost of flight tests, especially in periods of emergency. From the foregoing considerations, the current procedure of engine installation development by flight testing deserves careful economic consideration. If pre-flight tests may be devised for reducing the amount of flight time required, the provision of adequate facilities for conducting such tests will be justified.

Additional factors in favor of test-stand proving of powerplant installations are: the convenience of testing such a unit independently, without involving the many other details of the airplane; the effectiveness and economy of unit testing for development and changes on the installation; the speed with which an installation may be perfected, since a test stand may be run day and night; and the availability of the engine installation before the remainder of the airplane is ready for flight.

The Vega Aircraft Corp. has for several years recognized the advantages of the ground test stand for developing power units. For example, Vega-built cowl assemblies destined for shipment to England, to be used on the four-engined Armstrong-Whitworth "Ensign" transport airplane, were thoroughly proved on the stand before the design was finally released. As a result, the cowls worked

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 12, 1942.]

satisfactorily when installed in England, and are still giving good service. Another example of Vega test-stand activities is the novel test trailer which was built for the Unitwin powerplant installation. This test-stand trailer, complete with all the equipment necessary for testing, could be towed with a truck to any desired point where the atmospheric pressure and weather satisfied the conditions desired for a test. Such tests proved very useful in the development of the Unitwin.

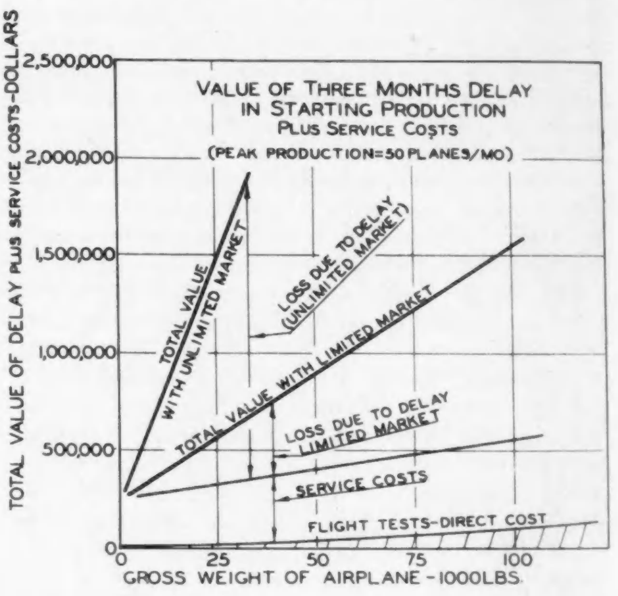
■ Some Vega Test Results

With experience and confidence resulting from such activities, the forementioned reasons were considered sufficient to justify a complete mockup and test-stand program for a Pratt and Whitney engine installation in a late-model airplane. A contract for a large number of airplanes was involved; the need for haste in starting production was quite apparent; hence, a program was outlined for determining as many flight characteristics of the engine installation as possible on a test stand. Preliminary experimental tests had already been conducted, such as powered wind-tunnel model tests, blower tests on the carburetor air-duct and oil-cooler installation, engine and propeller vibration tests, and so on. The engine installation was designed carefully for production, and the tooling and manufacturing divisions were anxious to proceed with construction. The next step was obviously to try the installation on a test stand in order to obtain the much-needed first approval of the design, rather than to accept a considerable delay, awaiting the airplane test flights.

The engine installation, Fig. 3, consisted briefly of an NACA-type cowl, with cowl flaps at the trailing edge; a long carburetor air duct on the top, with its entrance at the cowl nose; an oil cooler on the bottom, with entrance back near the cowl flaps; and an exhaust collector aft of the engine, with a single outlet on the lower, outboard quadrant of the cowl.

The test stand, Fig. 4, was designed and constructed of large-diameter steel pipe, which supported the engine installation in the ground or climbing attitude. A small test house was arranged alongside, with controls and instrumentation necessary for the program. This test stand was placed in a far corner of the Lockheed Air Terminal, aligned with the direction of the prevailing wind, and free from disturbing air currents from buildings. An organization of personnel was arranged whereby all test data were quickly correlated and distributed to various interested parties, and frequent conferences were called to discuss the results and decide upon necessary changes to the installation or to the test program.

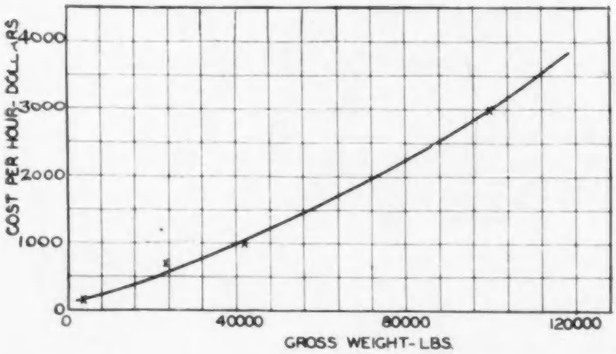
The conditions of operation for cooling tests consisted of continuous runs at 60% rated rpm and approximately 20% rated power for determining stabilized engine temperatures. Frequent short runs up to rated and take-off powers were made to determine the functioning of equipment under these conditions. Between these runs it was necessary to idle the engine for extended periods to cool down to normal temperatures. Under these severe ground-test conditions there were frequent delays due to service difficulties, such as fouled plugs, magneto and ignition troubles; but, in general, the tests ran off smoothly and results were obtained in a short time. Occasionally the wind direction and velocity caused operations to cease because side wind or excessive velocity gave erratic results, since the stand was not arranged to be pointed into the wind.



■ Fig. 2 – Combined cost of flight tests, service and potential loss due to production delay

A comparison of these ground-test data with ensuing flight-test results revealed some interesting points with regard to this engine installation, and also the method of test-stand operation as just described. The comparison served to strengthen our conviction that *pre-flight proving of engine installations can be done effectively if proper care is taken to simulate flight conditions.*

"Shakedown tests" to determine the functioning of various components of the installation, such as carburetor adjustment, propeller stop settings, oil-cooler doors and cowl flap actuation, supercharger and propeller governor controls and various accessories, agreed very well with airplane results. Minor adjustments and changes were, of course, found necessary on the stand, the knowledge of which avoided some delay on the first airplane. Similarly,



■ Fig. 1 – Direct cost of experimental flight testing

■ Fig. 3 – Engine installation on airplane

final adjustments were necessary on the airplane which were not apparent on the test stand.

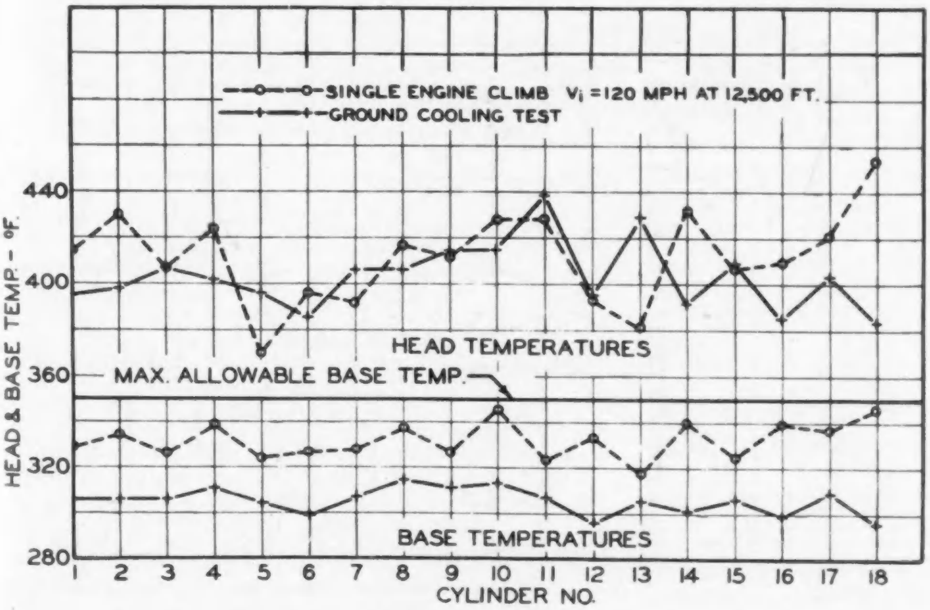
The most important question to be solved by the test-stand operation was that of engine and oil cooling. It was found, Fig. 5, that the stabilized temperatures from the 60% rpm tests agreed with the final flight test results at rated power, in so far as maximum cylinder temperatures were concerned. Cylinder base temperatures were slightly higher in flight, although still below the maximum allowable. Cylinder temperature distribution was different in flight than on the test stand, but no difficulty resulted from this condition. The method of testing was such that flight speeds and engine pressure differentials could not be obtained on the test stand by the propeller slipstream alone. It was therefore impossible to operate at rated power for extended periods, and these facts probably account for the discrepancies with flight data.

Carburetion had been the subject for much investigation before the test-stand runs. Duct tests had indicated alarming variations in velocity distribution across the carburetor entrance (as other manufacturers had discovered), but measurement of fuel metering differentials had indicated that the installation would function satisfactorily in spite of the velocity variations. Flight-test results bore out this contention for low-altitude conditions, Fig. 6. However, we experienced difficulty with carburetor enriching at high altitudes due to improper compensation, a characteristic which had not been tested in our preliminary work, but which has since been studied thoroughly by the carburetor manufacturer on ground tests as well as in flight, and the solution for this problem is now known.

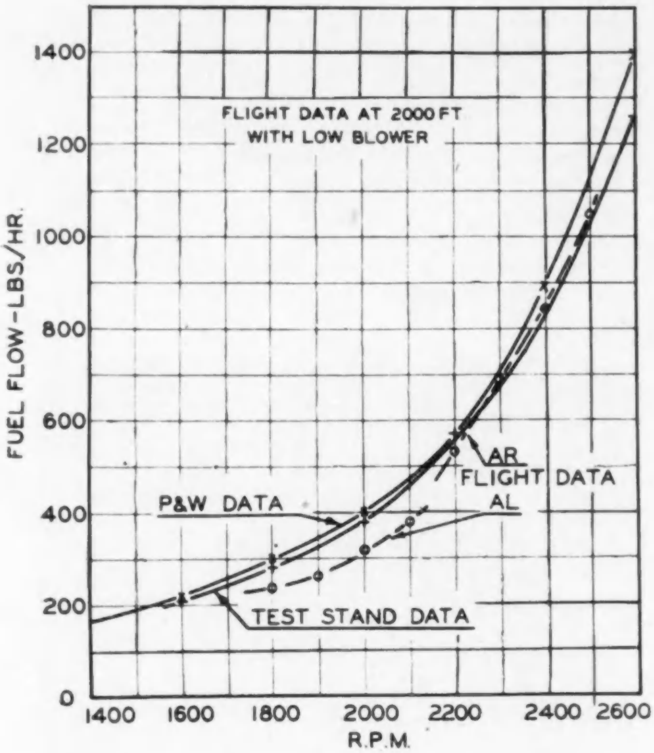
The carburetor preheat, as indicated by the test stand

■ Fig. 4 – Engine test stand

result, Fig. 7, was slightly unconservative, although in subsequent flight tests it was found that the heat rise was increased considerably by closing the cowl flaps, which had



■ Fig. 5 – Cylinder head and base temperatures



■ Fig. 6 – Carburetion tests

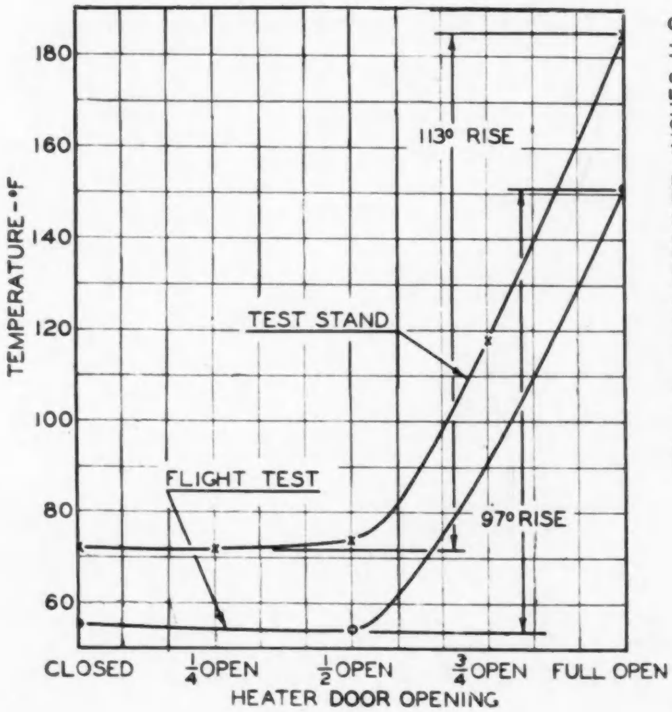
not been done on the original tests. In fact, with flaps closed, excessive preheat was finally obtained, and it was found necessary to reduce the hot air shroud volume to reach a satisfactory condition.

Exhaust back pressures, Fig. 8, which were obtained with high power conditions, agreed closely with values obtained in flight, both from the standpoint of maximum back pressure and distribution thereof.

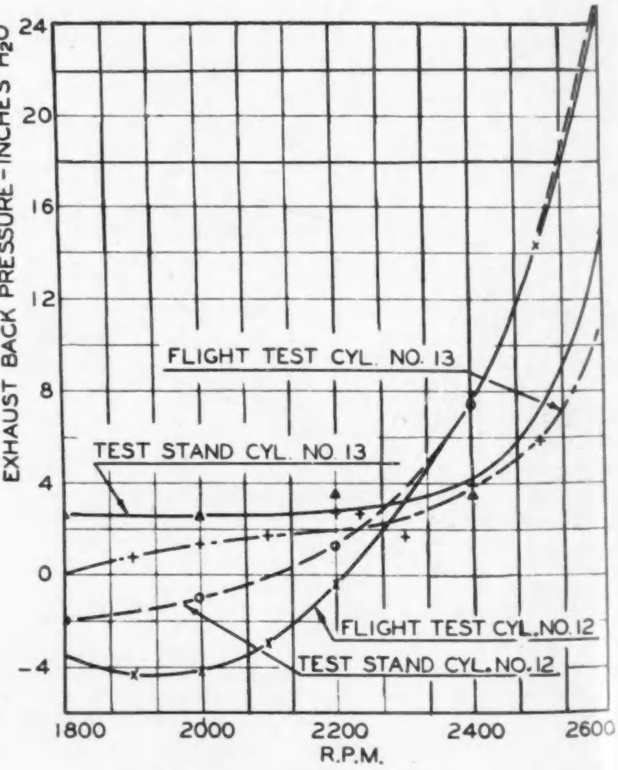
Accessory compartment temperatures, Table 1, were indicated qualitatively by the test-stand results, although slight differences were found in flight. The temperatures of items such as spark-plug elbows, flexible-pedestal mounts, and accessory-compartment equipment, were in general somewhat higher on the stand than in flight, probably due to lower differential pressures across the compartment.

Taxiing and ground run-up cooling characteristics agreed quite well with those of the prototype airplane. The major test-stand discovery in this regard was that amply large convection passages from the engine compartment were necessary in order to prevent overheating of spark-plug elbows after the engine was shut down. Unless sufficient convection was permitted, the heat content of the cylinder heads would quickly pass through the spark plugs and overheat the insulation of ignition wires, causing faulty operation on subsequent runs. The correction for this condition was to increase the number of cowl flaps near the top of the cowl, which arrangement worked satisfactorily on the airplane.

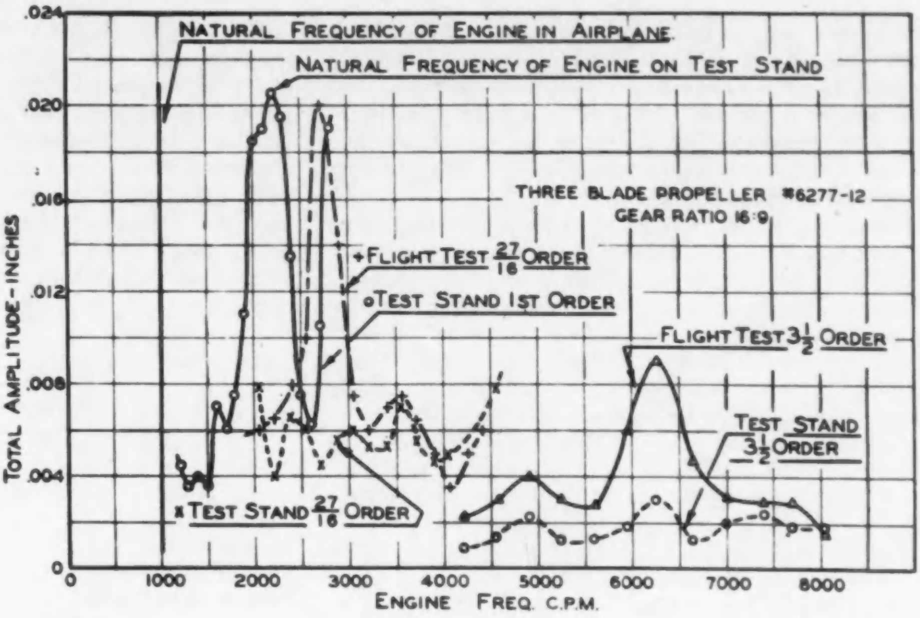
Engine and propeller vibration characteristics, Fig. 9, had been previously investigated on the propeller manufacturer's test stand. Due to the unusually wide planform of the propeller blade used on this airplane, rather high



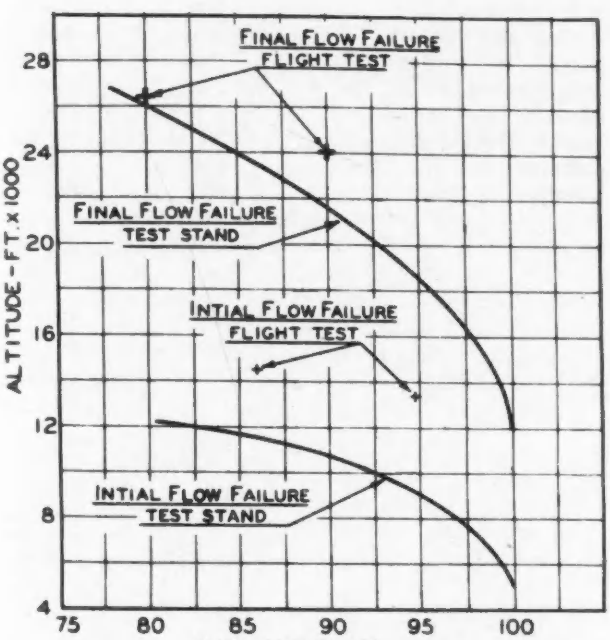
■ Fig. 7 – Carburetor heat rise



■ Fig. 8 – Exhaust manifold back pressures



■ Fig. 9 – Engine vibration amplitudes



■ Fig. 10 – Fuel flow under altitude conditions

excitation of the blade frequency due to fuselage interference, was to be expected (that is, engine 27/16ths order with the 16:9 gear ratio and three-blade propeller). However, since no means had been provided to excite the propeller-blade frequency on their test stand, such as a point of interference where the blades passed some large object, it was not surprising that little response to the 27/16ths order was recorded in these vibration tests. A resonance peak due to first-order engine motion was found, however, which agreed with a 27/16ths peak determined in flight tests, thus confirming this resonance point of the propeller-engine system; so it is probable that, if excitation of the propeller blade frequency had been provided on the stand, the results would have compared more favorably with flight test. Another order of vibration, the 3½ order, was fairly high on the airplane and somewhat lower, but of the same characteristics, on the test stand. The reason for this difference in amplitude may be that the test engine was mounted rigidly on the propeller-manufacturer's stand, whereas flexible pedestal mounts were used on the airplane, thus accounting for a change in amplitude without affecting the frequencies at which resonance peaks occurred.

Fuel flow tests were run with a bench-test rig separate from the engine, to determine the effect of altitude and fuel temperature conditions upon fuel pressures and flow, Fig. 10. The fuel pumping system consisted of the engine-driven pump and an auxiliary pump. An "initial failure" was defined as the pressure altitude at which the engine-driven pump, acting alone, dropped below 12 psi pressure. A "failure" was defined as the altitude at which the pressure of both the engine-driven pump and the auxiliary pump, acting in series, dropped below 12 psi. The results obtained on these bench tests were found to be slightly conservative, since higher critical altitudes were actually obtained in flight.

Table 1 – Corrected Engine Compartment and Accessory Temperatures, F

Test Condition	Max. Accessory Compartment Temperature	Max. Flexible Pedestal Temperature	Max. Fuel-Line Temperature	Max. Spark-Plug Temperature
Test Stand	155	219	128	247
Ground Cooling on Airplane	142	229	124	258
Maximum Observed Temperature During Flight	161	201	112	108

Numerous arrangements for flame suppression were tested on the stand, finally resulting in a simple means to eliminate flaming of the exhaust collector. This device was then tested in flight, and proved to be an effective method to prevent torching. The ability to try a multitude of different ideas on the stand undoubtedly accelerated the solution to this difficult problem.

A summary of the relationship between the foregoing test-stand and flight data, Table 2, shows that the test-stand results gave fairly good agreement with flight results, indicating flight characteristics in all cases where reasonable similitude was obtained, and that most examples of poor correlation were due to poor similitude conditions. *However, since the most important function of such tests is to indicate flight cooling characteristics, and since these conditions cannot be simulated accurately on a simple test stand, it may be concluded that, as an indication of flight cooling, the foregoing method of testing engine installations is, in general, not sufficiently reliable to warrant proceeding confidently with fabrication on the basis of such test results alone.* Ground cooling, and other tests where reasonable similitude is obtained, may be conducted in this manner with confidence in their reliability.

Philosophy of Pre-Flight Testing

The reliability of pre-flight testing may be improved by the logical procedure of analyzing the critical flight conditions carefully, separating the variables into factors of primary and secondary importance, devising a test procedure and facilities for reproducing as many of the factors as possible, estimating the degree of accuracy to be expected of the results, then judging whether or not the proposed test would be worth while. Model tests may sometimes be used to determine the relative importance of various factors. Several cases will now be considered as examples of the foregoing method of procedure.

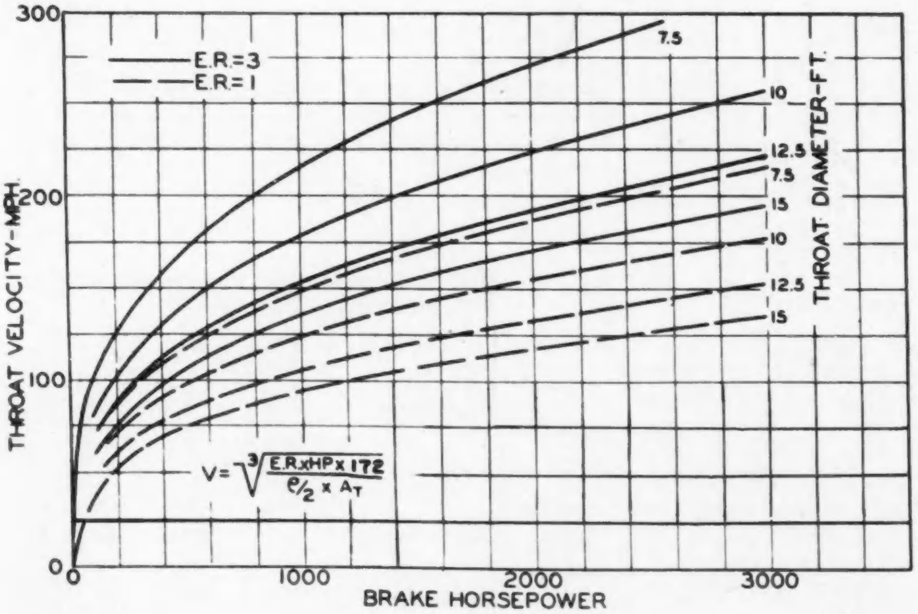
The primary factors affecting engine propeller vibration are as follows:

Given an engine with its characteristic internal excitations, mounted on a structure of the same stiffness as that of the airplane and driving a propeller of certain natural frequencies in its various modes, then the factors to be simulated for a vibration test are engine power and rpm conditions, aerodynamic excitation and damping, and propeller inflow velocity. Aerodynamic exciting forces (due to propeller tip-fuselage interference, for example) may be obtained on the test stand by reproducing the geometrical arrangement which causes the interference. Aerodynamic damping, which is proportional to the slope of the lift

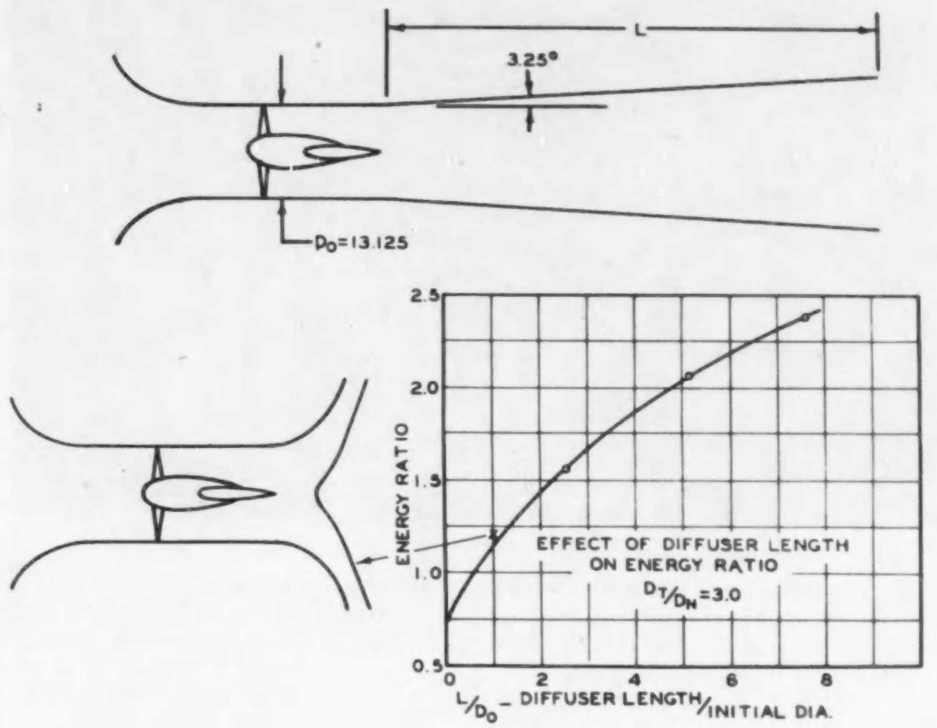
Table 2 – Correlation Between Test Stand and Flight Test Results

Item	Ground-Flight Correlation	Apparent Similitude
Shakedown test	Fair	Good
Ground Cooling	Good	Good
Cylinder-Base Cooling	Fair	Poor
Cylinder-Head Cooling	Good	Poor
Cylinder Temperature Distribution	Poor	Poor
Oil Cooling	Good	Poor
Low Altitude Carburetion	Good	Fair
Fuel Flow at Altitude	Fair	Fair
Exhaust Back Pressure	Good	Good
Vibration ($2^7/_{16}$th Order)	Poor	Poor
Propeller Stresses	Fair	Poor
Accessory Temperatures	Fair	Fair
High-Altitude Carburetion	No Test	No Test
Carburetor Heat Rise	Fair	Fair
Vibration ($3\frac{1}{2}$ Order)	Fair	Poor

curve of the blade elements, requires that the blades be unstalled and at the correct pitch setting, in order to approximate flight conditions. Inflow velocities affect the angles of attack of blade elements and the strength of the

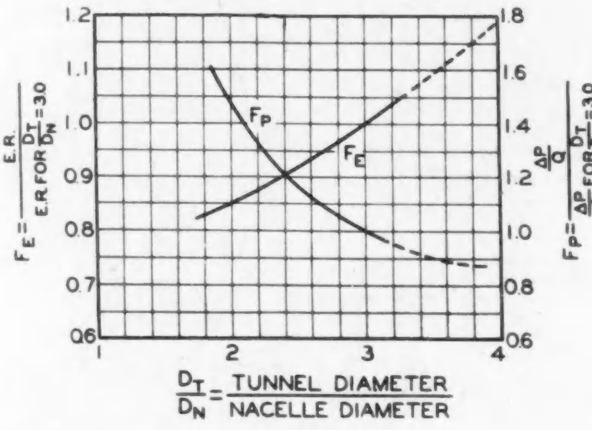


■ Fig. 11 – Wind-tunnel throat velocity versus power, throat diameter, and energy ratio



■ Fig. 12 – Energy ratio versus diffuser length

excitation at interference points. It therefore appears that, if vibration data are to be obtained with pre-flight tests, flight velocities as well as accurate reproduction of interferences must be provided in order to insure reliable engine



■ Fig. 13 – Nacelle diameter effect on energy ratio and pressure differential across the cowl

amplitudes and propeller stresses. This condition suggests either an airplane of the general configuration in question, or a large wind tunnel with a powerful auxiliary motor, as the necessary test equipment for accurate results. If such facilities are not available, certain preliminary test data, such as resonance frequencies, may be obtained on a simple test stand, but the amplitudes resulting from such tests will be unreliable in general.

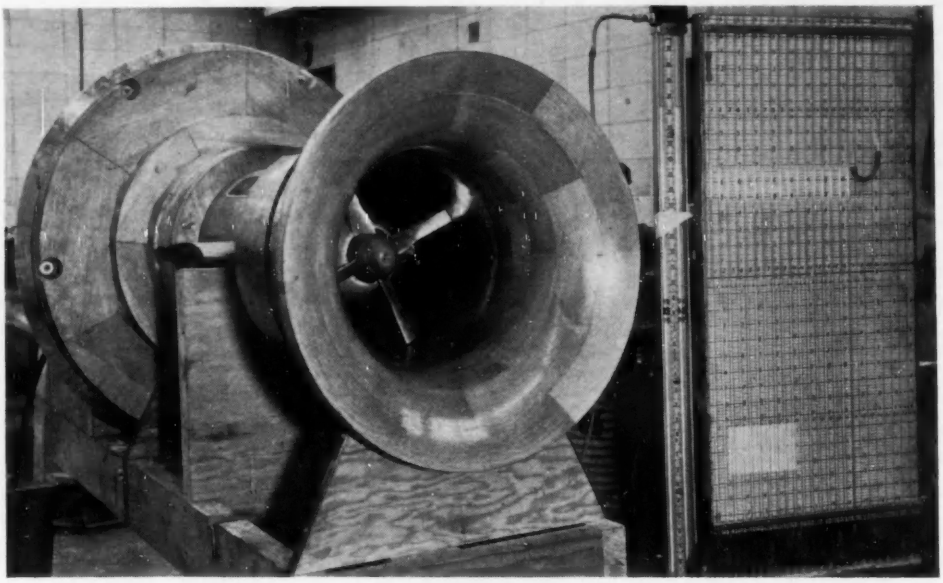
Considering next the similitude conditions which should be satisfied for an accelerated service test on an engine installation (performed under transport operation conditions, with 2-hr cycles consisting of take-off power for 1 min, rated power for 5 min, cruise for 100 min, idle for 10 min, and so on) the primary factors will probably be engine power conditions, flight velocities and pressure differentials, fuel consumption and vibration characteristics. The facilities required for such tests are the same as for vibration tests just discussed, and for engine cooling, to be described more fully in the following paragraphs.

The primary factors affecting engine cooling, oil cooling and compartment temperature for a given nacelle arrangement, are usually engine power, fuel consumption, airplane and slipstream velocity, and pressure differential across the engine or cooler. Secondary factors are usually altitude conditions, angle of attack, slipstream rotation, carburetor preheat, and so on. Several means for reproducing these conditions are as follows:

■ Test Facilities

1. An airplane "flying test stand" may be used, with the engine installation under investigation serving as motive power. Although pre-flight tests may be conducted with such equipment, the time and expense involved (see Fig. 1) in making carefully controlled flight tests of this type should be considered in comparison with ground-testing methods.

2. One method for creating higher velocities on a ground test stand is to place the engine and propeller installation



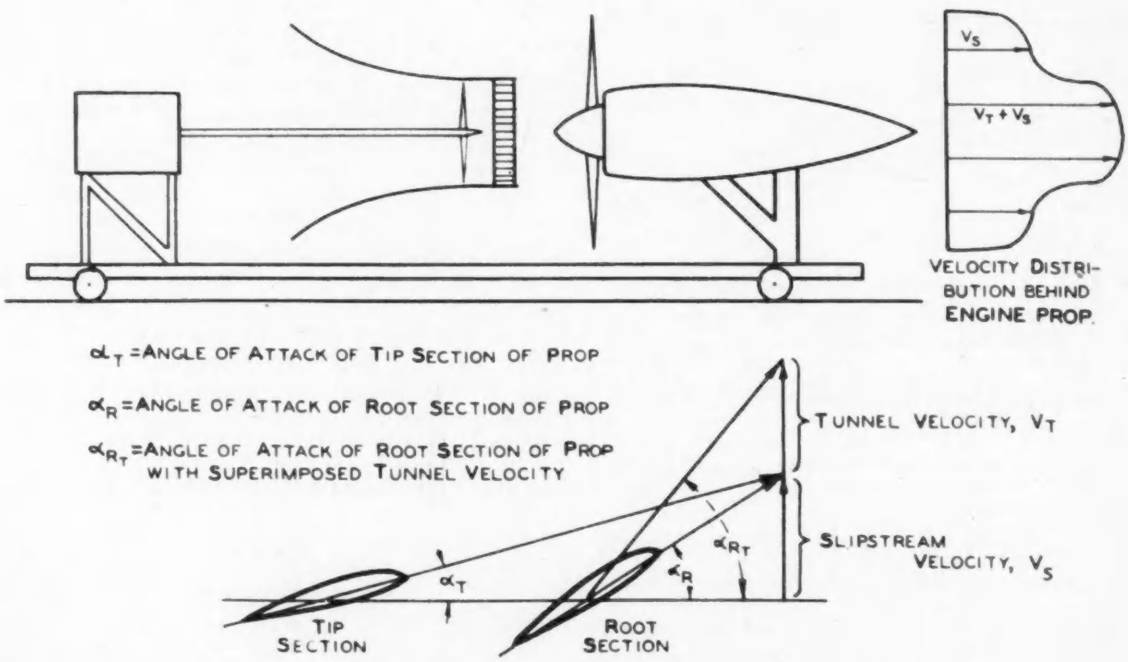
■ Fig. 14 – Small model of wind tunnel-nacelle propeller arrangement

at the throat of an open return-type wind tunnel, relying upon the engine power itself to provide the sole motive power for the tunnel. The throat velocity V in mph, Fig. 11, may be represented by the equation:

$$V = \sqrt[3]{\frac{\text{E.R.} \times \text{Bhp} \times 172}{\rho/2\, A_t}}$$

where E. R. = Energy ratio of tunnel with test nacelle in place.
Bhp = Brake horsepower of engine.
ρ = Mass density of air – slugs/cu ft
A_t = Throat area of tunnel – sq ft

Test values of energy ratios obtainable with various lengths of diffuser are indicated in Fig. 12, and the corresponding performance may then be determined from the chart. It is assumed that the propeller diameter is nearly equal to the throat diameter, with a small tip clearance for the blades. A reduction in the ratio of throat diameter to nacelle diameter will affect the energy ratio slightly, Fig. 13, as well as the pressure differential across the nacelle. It appears from test results that the minimum D_t/D_n ratio is approximately 4.0 to simulate free air conditions of pressure differential across the engine. However, model tests (Fig.

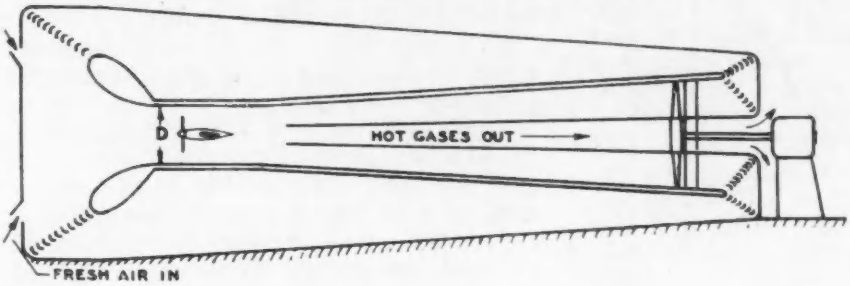


■ Fig. 15 – Auxiliary fan for producing high velocity over nacelle

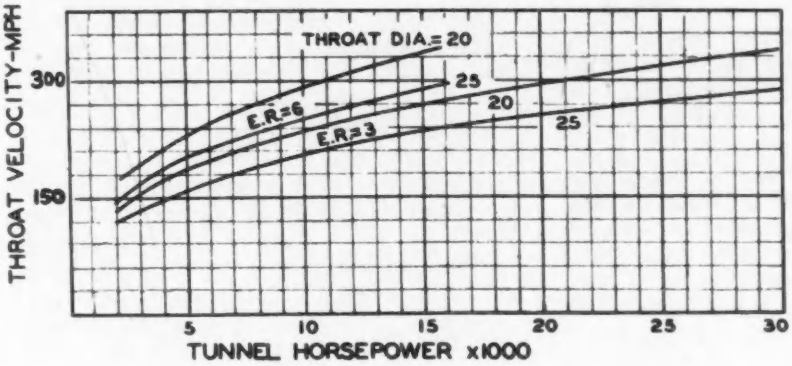
14) of a proposed wind tunnel-nacelle arrangement may be used to determine the wall effect upon the pressures, and thereafter the cooling data may be interpreted to give free air results for D_t/D_n ratios less than 4.

3. Another means for creating flight velocities over the test installation is to provide an auxiliary propeller or fan some distance ahead of the test propeller, Fig. 15, which will create a blast of air of a diameter slightly greater than the nacelle. The outer portion of the test propeller will operate under static thrust conditions, whereas the center portion will receive the auxiliary fan slipstream and, due to the increased velocity, the root blade sections will assume a more negative angle of attack, tending to reduce the effect of the auxiliary blast by a questionable amount. This complication to the test propeller blade angles, together with the difficulty of accurately measuring the effective velocity acting on the nacelle, make the true flow conditions very uncertain. For these reasons, the method is of doubtful value for engine testing.

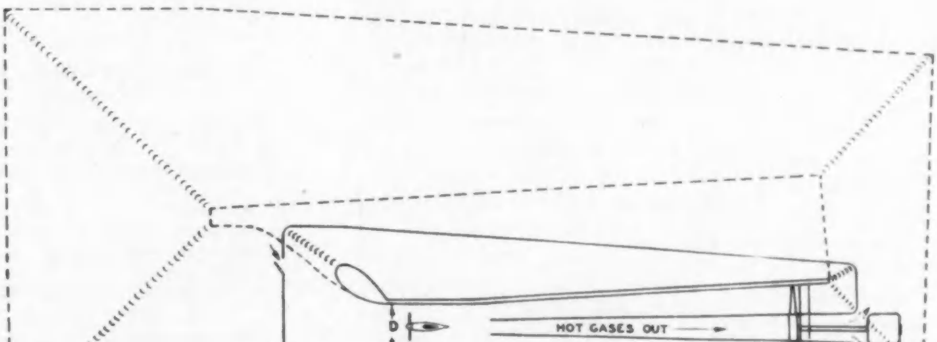
4. A final proposal for increasing the test velocities is to provide an adequately powered wind tunnel, preferably of the closed-return type, with the test installation at the throat, supplying its share of the total power. The most efficient arrangement of this type in current use is the single return duct, with guide vanes at the corners, and a long, gentle expansion between the test section and the



■ Fig. 16 – Compact form of closed-return wind tunnel



■ Fig. 17 – Comparison of size of normal single-return wind tunnel with annular-return type



first corner. Although very suitable for test purposes, one disadvantage of this type of tunnel is its tremendous size to accommodate a throat diameter large enough for engine and propeller testing.

As a means to reduce the size and cost of the closed return wind tunnel, an Eiffel-type tunnel of efficient form may be provided, Fig. 16, enclosed within a building slightly larger than the tunnel itself, the walls of the building forming an expanding annular return passage for the flow. This would present a very compact arrangement of approximately one-third the volume of a normal single closed-return wind tunnel having the same energy ratio. See Fig. 17. Further reductions in overall dimensions may be possible by the use of boundary-layer control on the inside and outside of the conical section, permitting its length to be reduced by reason of a wider cone angle, without sacrificing tunnel efficiency.

When altitude conditions are a primary factor in engine cooling, it may become necessary to simulate altitude pressures and temperatures within the tunnel. In such cases, the foregoing proposal is especially advantageous, for the following reasons: With dished ends, its form is much simpler to adapt to a compression or evacuation chamber. The quantity of steel required will be approximately one-third as much as for a single closed-return type. The evacuating equipment and likewise the refrigerating equipment, will be less expensive because of the lower volume within the shell. The reduced surface area will permit a further reduction in refrigeration costs, from the standpoint of refrigerating capacity and insulation costs. The tunnel power required to obtain a given speed may also be reduced when operating at low pressures. For example, under pressure conditions corresponding to 40,000-ft altitude (approximately 1/4 atmosphere), the throat velocities will be 1.6 times as great as values given in Fig. 16.

It should be mentioned that the methods for ground proving of engines described herein apply as well to radial as to in-line engines, either liquid or air cooled. Likewise, small engines may be tested by these methods, in which case the test equipment involved will be reduced in scale and cost, in agreement with the size of engine and the value of the tests.

From the foregoing methods for creating flight conditions for the powerplant, or other methods which may be devised, a wide selection of possibilities will be available to satisfy economic and time considerations, as well as test conditions. With such facilities, the following characteristics may be proved on the test stand before flight testing is begun:

- Shakedown tests
- Engine cooling
- Oil cooling
- Fuel flow and vapor lock
- Carburetor icing
- Exhaust back pressures
- Turbo-supercharger functioning
- Exhaust jets
- Engine detonation under service conditions
- Measurement of drag of engine installation
- Propeller and engine vibration
- Serviceability of installation

If this work is done effectively, it will permit a welcome reduction in the amount of flight testing required to approve a new model, as well as a saving in cost to the manufacturer.

Final Remarks

In consideration of the numerous ground tests which may be conducted, a careful selection of test equipment to serve best all the needs is required, and a re-evaluation of the total benefit of such equipment, combining its usefulness for experimental development with that of service testing. A comparison of the various types of wind-tunnel previously discussed, Table 3, indicates that the closed-return type with adequate power will give satisfactory similitude for all major test conditions, such as engine cooling, vibration, carburetion, service test, and so on. For large engines, the tunnel throat should be approximately 25 ft in diameter in order to permit angle-of-attack variation and accurate vibration and service tests. Assuming an atmospheric tunnel of energy ratio 3.0, the power required for a speed of 250 mph will be 16,000 hp. (With an energy ratio of 6.0, only 8000 hp are required.)

An analysis of the normal operating costs for such a tunnel (including depreciation and interest on the investment) shows that such tests may be conducted more economically than even the direct costs of flight testing, Fig. 1, the advantage becoming increasingly great with the size of the airplane involved. Another factor is the cost of tying up a valuable prototype airplane for detailed tests and changes. With prototypes valued at up to several million dollars, this factor becomes one of great importance.

The foregoing analysis seems favorable for the engine test wind tunnel but, even more convincing, is the argument presented by a statement of the combined value of flight tests, losses due to production delays, and service costs, Fig. 2. If some difficult engine installation problem delayed production for several months, the resulting loss to the company might exceed the cost of an adequate wind tunnel. Likewise, if some undiscovered service difficulty arose after a large number of airplanes were delivered, the cost of rectifying this condition in the field might amply justify a considerable investment in test facilities of the type described. These may be unusual examples, but such incidents may be cited in the history of almost every large airplane manufacturer, where the pressure to start production has caused a disproportionate expenditure due to a long flight test program or unforeseen service difficulties.

It is safe to say that, on the average, development and service tests may be accomplished in the wind tunnel in a shorter elapsed time, and at a fraction of the total cost of such tests conducted on the airplane. The logical conclu-

Table 3 – Comparison of Various Types of Engine Test Equipment

	Test Condition	Simple Venturi	Auxiliary Jet	Closed-Return, Auxiliary Power
Similitude	Airspeed	Climb	Any	Any
	Airflow	Fair	Poor	Good
	Vibration Interference	None	Fair	Good
	Cooling Air Temperature Control	None	None	Good
Usefulness	Experimental Development	Good	Poor	Very Good
	Vibration Tests	Poor	Fair	Good
	Service Tests	Fair	Poor	Good
	Relative Cost	2	1	3
	General Preference	2	3	1

sion to the foregoing arguments is that every airplane manufacturer should own, or have access to, an adequate engine test wind tunnel.

Bibliography

T. G. Hill and A. A. Joyce: "Ground Testing Power Plants," *Aviation,* October, 1941, pp. 64-65, 150, 152.

F. C. Mock: "Aircraft Carburetor Airscoops and Their Effect of Fuel-Air Metering in Flight," SAE Transactions, Vol. 50, March, 1942, pp. 102-124.

W. C. Rockefeller: "Powerplant Wind-Tunnel Testing," presented before the Annual Meeting of the Society, Jan. 7, 1941.

W. F. Durand: "Aerodynamic Theory," Vol. III.

Louis Massuger: "Aero Engines and Their Laboratory Testing under Flight Conditions," *Journal of the Aeronautical Sciences,* Vol. 7, September, 1940, pp. 481-482.

F. L. Wattendorf: "The Efficiency of Return-Flow Wind Tunnels," Tsing Hua University Science Report, July, 1936.

G. N. Patterson: "Modern Diffuser Design," Aircraft Engineering, Vol. 10, September, 1938, pp. 267-273.

Possibilities of Increasing Automotive-Engine Output

THE problem of obtaining higher output from automotive engines may be simplified to one of possible improvements in thermal efficiency and volumetric efficiency. Thermal efficiency of gasoline engines may be improved by: increased compression ratio, high-octane fuels, lower charge temperature, lower chamber temperature, and decreased flame travel. In diesel engines thermal efficiency may be improved by rapid and complete combustion, improved mixing, improved injection, improved scavenging, and high chamber temperature. Volumetric efficiency of both engine types can be increased by less intake restriction, lower charge temperature, lower exhaust back-pressure, supercharging, and two-stroke cycle.

■ Thermal Efficiency

Thermal efficiency is discussed first since, without control of heat distribution within the engine, it is useless to attempt to improve output by increasing the rate of energy input. It is waste heat, not useful heat, that is destructive to the internal-combustion engine. There is an old saying that it is easy enough to get fuel to go into an engine, but it is the efficiency with which it is used that determines whether the engine is to be a pig or a horse.

By far the most important factor which influences thermal efficiency is the compression ratio. This has long been understood, but it has not always been appreciated that there is a limit to compression beyond which it does not pay to go. Tests carried out on an automobile engine show that, in this case, the maximum output is obtained with a compression ratio of between 9 and 10:1.

The reason for this result seems to be that, with increasing maximum pressures, engine friction is going up at a higher rate than indicated efficiency, when the actual working fluid and heat suppression during the high-temperature part of the cycle are considered. The limiting compression ratio is somewhat higher in very small cylinders which show their highest output at compression ratios in the neighborhood of 12:1.

Another factor is that the volumetric efficiency tends to fall off slightly with increasing compression ratios. The exhaust gas which is held in the compression space at the end of the exhaust stroke cools and contracts during the intake stroke. When the compression space volume is decreased to afford an increase in compression ratio, the usefulness of this effect tends to diminish.

The fact is that the gasoline engine has no desire to emulate the diesel engine and to use compression ratios of the order of 16:1. On the contrary, the diesel would like to get rid of a few ratios if it could start from cold on any lower compression temperature.

■ Detonation

Any attempt to increase the compression ratio of a well-designed engine running on its normal fuel will, of course, be halted by the onset of detonation. The highest useful compression ratio depends upon the stability or knock resistance of the fuel under the working conditions, and these conditions in turn are a matter of engine design and maintenance.

Consider for a moment the treatment meted out to the incoming fuel before, breaking ranks, it finally decides to burn uncontrolled. The charge is heated in the manifold and by contact with residual exhaust gas. It is heated by compression and by radiant heat from the head of the exhaust valve and piston crown. Apparently, at this time, slow combustion commences, using up fuel and adding still more heat to the mixture. Finally, the flame further compresses the unburnt charge and, encouraged by the self-ignition of drops of oil thrown from the piston, it finally decides to ignite for itself.

It seems clear, therefore, that the highest useful compression ratio of an engine depends not only on the cylinder bore, engine speed, and so on, but upon everything which has to do with the temperature of the portion of the charge last to burn, just prior to its ignition. With careful design and using special racing fuels with a high alcohol content, it is possible to use compression ratios as high as 10:1 and, incidentally, to obtain outputs up to 1.25 hp per cu in. at 6000 rpm, without supercharging.

Such fuels are not, of course, available for the road, rail, and military engines which we are considering and, with a fuel specification of 70 octane minimum rating, a compression ratio of 6:1 is the usual limit. Once the engine has been designed and built, the compression ratio is settled by trial, allowing for carbon build-up and loss of compression in service, while permitting a knock of medium intensity in the lower end of the speed range.

Also, it is not possible to use the small bores (under 3 in.) common in racing engines, or their aluminum heads and pistons. Nonetheless, much can be done by careful design, especially of the pocket which contains the portion of the charge last to burn, and by the use of salt-cooled exhaust valves.

Rapid combustion is important from the point of view

of thermal efficiency, but presents no difficulty with the turbulence which is always present in the combustion chamber. Scavenging is important, in that too much exhaust dilution tends to slow down the rate of burning of the charge.

■ Thermal Efficiency in Diesel Engines

Turning now to the problem of improving output by increasing thermal efficiency in diesel engines, we are faced with an entirely different set of factors. First, the diesel engine already has a comparatively high thermal efficiency on account of its high compression ratio, necessary to start from cold; its inability to work with other than comparatively lean mixture ratios and, in the automotive type at any rate, its ability to work on a mixed cycle far closer to the Otto than the diesel cycle in efficiency. Second, fuel quality has very little influence on its thermal efficiency.

Comparing the relative indicated thermal efficiency of gasoline and diesel engines, it is seen that the diesel would make a poor showing in spite of its high compression ratio and its 25% excess air at full load were it not possible to work small diesel engines on a mixed cycle with a maximum explosion pressure about 1.75 times the compression pressure and, by so doing, to regain some 70% of the efficiency lost by going to the diesel (constant-pressure) cycle. The mechanical efficiency of the diesel engine is lower than that of the gasoline engine but, for automotive work, this is more than offset by the improved thermal efficiency of the diesel at light loads, where the mixture ratio may be as lean as 100 lb of air per lb of fuel.

Referring now to the fuel, there is no quality in diesel fuel which has anything like the influence on output as has the antiknock quality of gasoline fuel. It is true that cetane rating and possibly volatility have some influence on maximum output but, compared to the octane rating of a gasoline fuel, their effect is unimportant, however valuable they may be from other points of view.

This does not mean to say that there is no room for improvement in the thermal efficiency of diesel engines; the fact is that most production automotive engines do not work at or near a maximum economy at full load.

Automotive diesel engines are run at full load on the hump of the curve in the interests of maximum output. It is therefore for the designer to provide the simplest and most compact combustion chamber which will, by coordinating fuel jets and air movement, afford the maximum output without disproportionate fuel input. While very little is known about the combustion process, enough experience has been obtained to design chambers which afford a satisfactory degree of smoothness through combustion control, and an economy which approaches that of large stationary diesel engines, with close to 80% air use. The final details must still be settled by trial, for economy on a single-cylinder engine.

Scavenging is important in obtaining high output, not only to burn the greatest quantity of fuel per cycle, but because combustion, comparatively slow at best, tends to be slowed down by exhaust-gas dilution, with obvious ill effects on economy and output.

■ Volumetric Efficiency

The volumetric efficiency of any well-designed and properly installed gasoline or diesel engine is already high and leaves but little room for improvement. Ruling out the use of alcohol fuels by means of which volumetric efficiency can be increased by about 8%, all that one can do is to check carefully the factors which control charge density. These factors include the heat input to the intake manifold; area of venturi, intake passages, and valve ports; shape of intake port, valve head, and valve seat; valve timing, intake-lift curve and assurance that this curve is followed, by use of hydraulic lifters, correct design of cam, and spring, and so on; clearance about intake valve head in chamber; cooling of chamber, compression ratio, and exhaust back-pressure.

In the type of engine that we are considering, the demands of easy starting, smooth idling, and high output at low speeds fix a definite limit to improvement in these directions. Thus, with regard to intake velocities, we seem to be limited to something like the following, at 1000 rpm:

	Velocity, fps
Carburetor venturi	150
Riser, updraft	90
downdraft	80
Manifold branches	65
Intake valve	90
Exhaust valve	125

Excerpts from the paper: "Automotive Engines of High Output," by P. E. Biggar, General Motors Truck & Coach Division, Yellow Truck & Coach Mfg. Co., presented at a meeting of the Southwest Group of the Society, Oklahoma City, Okla., Jan. 9, 1942.

Air Flow Through Intake Valves

(*Concluded from page* 220)

effective area of the intake passage is large for a greater fraction of the valve-opening period.

The valve lift is limited in an engine by the necessity of getting the valve open and shut quickly. The designer is, as usual, faced with the necessity of making a compromise. In automobile and aircraft engines the maximum valve lift is in the neighborhood of one-fifth of the valve-head diameter.

The flow coefficient at maximum L/D is especially important because a very large fraction of the charge enters the cylinder with the valve at nearly full lift. The opening is biggest then and remains so for some time, and the pressure drop across the valve is high.

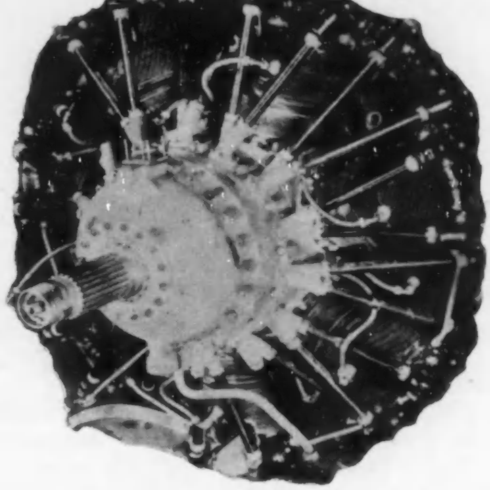
■ Conclusions

From the results recorded here, certain general conclusions may be drawn:

1. All corners should be rounded.
2. The fillet between valve stem and valve head should not be too large.
3. Port elbows should be laid out with a generous radius, and should not have any abrupt changes in passage area.
4. Cylinder walls can, and should, be formed and located so as to assist the flow.
5. Work done on valve and port combinations to improve their flow coefficients may be confined to lifts near the maximum which will be used.
6. Results of tests made with a small pressure drop across the valve may be applied through the working range of pressure drops unless there is reason to expect a considerable pressure recovery in the expanding part of the passage between the valve and its seat.

Some Notes on Design Features of THE MITSUBISHI KINSEI ENGINE

by W. G. OVENS
Staff Engineer, Wright Aeronautical Corp.



■ Fig. 1 – Mitsubishi Kinsei engine similar to that described herein (left front view)

RESULTS of a study of a Mitsubishi Kinsei engine taken from a crashed Japanese plane are presented in this paper. The engine was made available to the author by the Experimental Engineering Section of the Army Air Force Materiel Center, Wright Field, O. The following are among the conclusions drawn by Mr. Ovens:

The engine "is undoubtedly a highly dependable, even though not highly developed, piece of equipment, probably produced under time and tooling limitations which we would consider nearly impossible."

The designers "did a very ingenious job of combining what they apparently believed to be the most desirable features of a number of foreign engines – proved features all – into a composite design that 'has to work the first time,' and probably did."

■ ■ ■

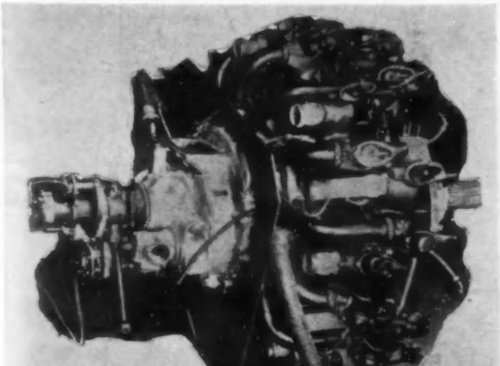
THE AUTHOR: W. G. OVENS, (M '35), staff engineer, Wright Aeronautical Corp., at present is head of the 14-cyl Cyclone Project Engineering Group, and since 1935, has worked on the Wright two-row 14-cyl aircraft engine development. He has been with Wright since 1929. While holding down varied jobs in machine shops and garages from 1922-1929, Mr. Ovens studied mechanical engineering for two years at Lehigh University.

THE condition of the only physical engine available for study and the data readily available can form the basis for only a very meager report. The study has, however, been an interesting one and the results are recorded for what value they may have. The design comments are, of necessity, of a general nature – much the same as those which would be made on the preliminary layout of a new design. For the convenience of many of us who habitually think in terms of English units, these units are used even though a large portion of the work is apparently based on the metric system. As a result, the numerical data are approximate conversion figures in the hope that these figures will best serve the purpose intended.

The inspection indicates to the writer two possible conclusions which are presented herewith:

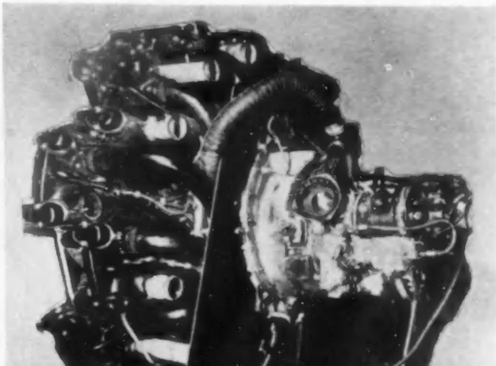
1. That the group responsible for the design did a very

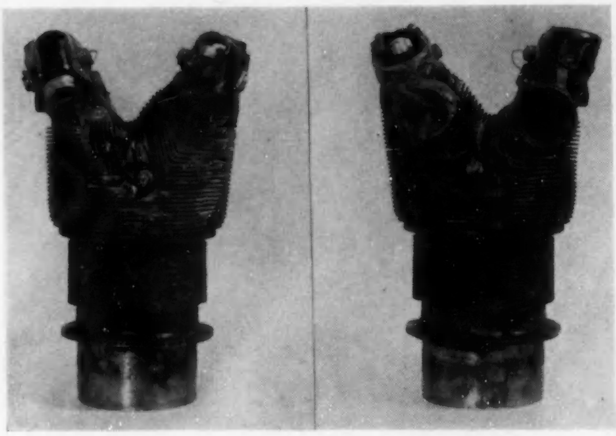
[This paper was presented at a joint Meeting of the Detroit Section of the Society and the Engineering Society of Detroit, Detroit, Mich., June 8, 1942; it was scheduled originally for presentation at the 1942 Semi-Annual Meeting of the Society, which was ruled out because of transportation priorities.]



■ Fig. 2 (at left) – Right rear view of engine shown in Fig. 1

■ Fig. 3 (at right) – Left rear view of engine Shown in Fig. 1





Front View Rear View

■ Fig. 4 – Complete cylinder assembly – Front and rear bank cylinders appear to be identical except for push-rod angle

ingenious job of combining what they apparently believed to be the most desirable features of a number of products of foreign manufacture – proved features all. These features are built into a composite design of the sort that "has to work the first time" – and probably did.

2. That manufacturing methods and equipment of manufacturers whose features were appropriated were probably used to produce parts of quality comparable to the originals; and that the available "heavy-industry" equipment probably influenced both the design and finished parts which are peculiar to this engine. In short, I am trying to convey the idea that this is undoubtedly a highly dependable, even though not highly developed, piece of equipment; and that it was probably produced under time and tooling limitations which we would consider nearly impossible.

The report is made possible by the graciousness of the Experimental Engineering Section of the Army Air Force Materiel Center, Wright Field, O., in making the engine available for study. The spirit of cooperation of the personnel of that Section in the disclosure of their findings and in the discussion of the subject is also gratefully acknowledged. Much of the detail investigation was carried out with the excellent assistance of the Materials Laboratory and other engineering personnel at the Cincinnati, O., plant of the Wright Aeronautical Corp.

■ General Data and Discussion

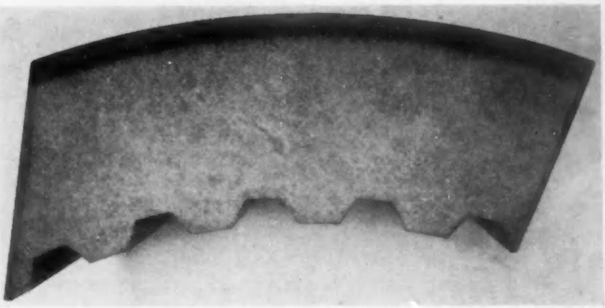
Type	Radial aircooled
Cylinders	14, 5.5-in. bore x 5.92-in. stroke
Cylinder arrangement	Two radial banks of 7
Engine Diameter	47 in. approximately
Piston Area	332 sq in.
Displacement	1970 cu in.
Compression Ratio	6.6:1
Supercharger	Centrifugal, 9.62-in. diameter impeller
Supercharger drive	8.48 x crankshaft

Performance *estimates* on 95- to 100-octane fuel based on American standards of service life.

Maximum Cruise	600-650 hp – 2000 rpm
Rated	850 hp – 2250 rpm to 8000 ft
Military Rated & Take-off	1050 hp – 2500 rpm to 5500 ft

General engine condition (before it crashed) was very good. See Figs. 1, 2, and 3. Evidence would tend to indicate that it had been operated for only a short period since overhaul but that that operation had been satisfactory. Pistons, cylinder barrels, valves, rods, reduction gear, and so on, which are available for inspection are excellent. Parts of the supercharger and accessory drive are mutilated badly enough to make any comments on this section invalid. There is, however, some indication that an impeller thrust bearing failure may have taken place prior to the crash, but that the shaft continued to operate against the spherical face.

Cooling provision is the only weak spot from a serviceability standpoint, in the writer's opinion. Potential output is probably limited to approximately 0.5 hp per cu in. by this feature. A rather rough estimate places the cooling area per cylinder at somewhat less than 1000 sq in. Fig. 4 shows the complete cylinder assembly. The baffle pressure drop to keep combustion-chamber temperatures below the detonation point would almost certainly make American aircraft designers very unhappy.



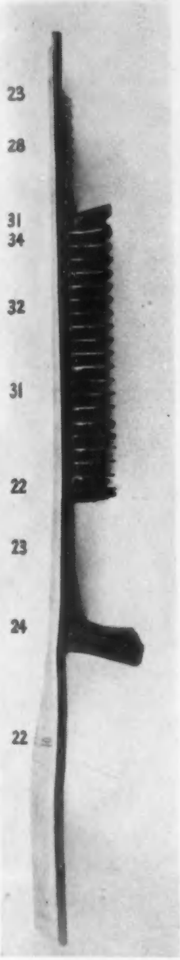
■ Fig. 5 – Section of crankpin showing internal splines and case-hardened journal surface

■ Fig. 6 – Section of starter and accessory driveshaft showing case-hardened journal surface on material which is hardened throughout when applied to propeller shaft

Given the cooling limitations just mentioned, it is believed that the remainder of the engine is very conservatively designed. With these removed it is probable that master-rod bearing lining cracking would soon develop, not because of excessive bearing loading but because of flexure of the rod hub. The carburized crankpins can be expected to aid bearing performance, and the lubricating means is presumably adequate since it is as used by another manufacturer with good results reported.

Some of the materials used in the Kinsei engine are of interest. They indicate that, at least at the time when this engine was built, there were adequate supplies of nickel, cadmium, chromium, cobalt, copper, molybdenum, and tungsten.

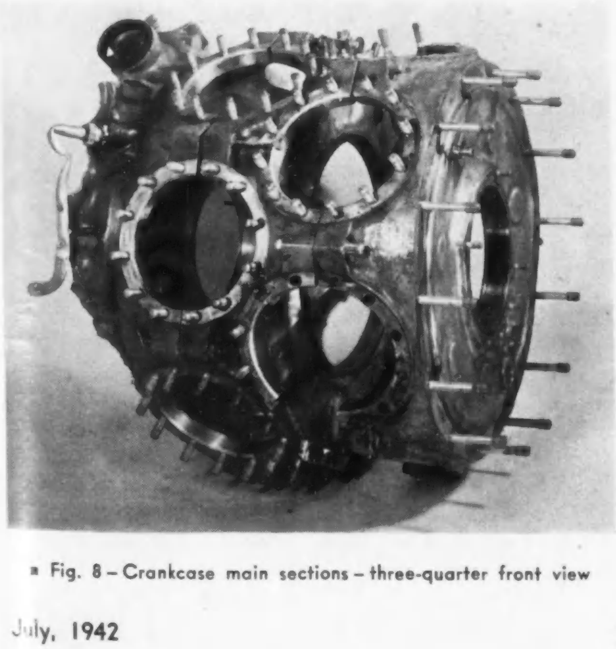
The one magnesium alloy found varies somewhat from American standard alloys in that it contains 4.6% aluminum, 2.6% zinc, and 0.28% manganese in addition to magnesium. It will be noted that this alloy is similar to AMS 4424 except that the aluminum content is low.



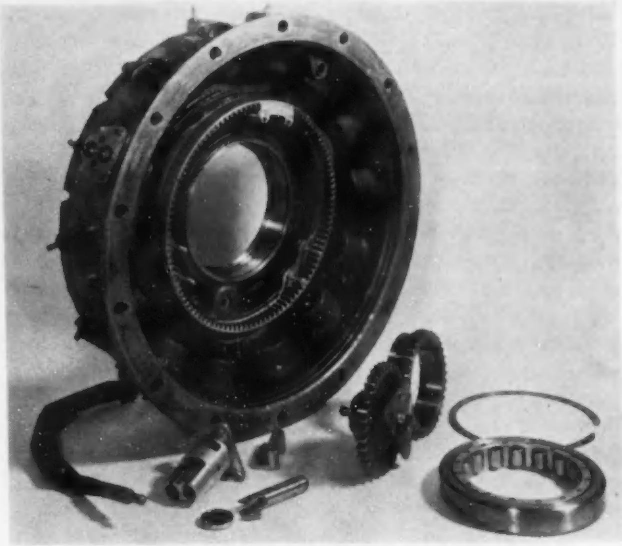
In the aluminum alloys found, 17S is used for many parts such as main crankcase, tappet guides, piston-pin plugs, and so on. For special purposes such as pistons, cylinder heads, and supercharger front housing, an alloy containing 3.93% copper, 1.37% magnesium, and 1.67% nickel is used either cast or forged.

An all-purpose steel, either case-hardened or hardened throughout, is used for connecting rods, crankshaft, valve rockers, and so on. Fig. 5 shows a section of crankpin. It contains approximately 1.5% chromium, 3.5 to 4.5% nickel, 0.3 to 0.4% molybdenum, 0.35 to 0.5% manganese, and varying small quantities of silicon and copper apparently as impurities. Carbon content is varied as required. The same steel with the molybdenum reduced and 0.5 to 0.9% tungsten and 0.2 to 0.4% cobalt added is used in the propeller shaft and in the starter and accessory driveshaft. The latter part is case hardened as shown in Fig. 6. It is suggested that this may well be a compromise for making the best possible use of the available scrap materials. Propeller reduction gears, cam, and knuckle pins are carburizing 4.5% nickel steel plus approximately 0.8% chromium. Reduction-gear pinions vary from this composition in the addition of 0.4% molybdenum. Nitriding is used only in the cylinder barrel (Fig. 7). The steel conforms very closely to AMS 6470. Nitride depth is 0.010 and 0.020 in. in two barrels cut. Core hardness varied from Rockwell C 22 to 34 in one specimen. Magnetic inspection of all steel parts illustrated showed acceptable material.

■ Fig. 7 – Section of cylinder barrel showing variation in Rockwell C core hardness of the only nitrided part found in the engine



■ Fig. 8 – Crankcase main sections – three-quarter front view



■ Fig. 9 – Intermediate crankcase front section and valve gear with a pair of tappets, cam, and cam bearing in place

Detached parts:
Cam Retainer
Push rod and housing
Cam drive gears
Tappet guide
Bearing race lock ring
Tappet and pin
Tappet roller
Front main bearing which supports reduction driving gear

Plating is used quite extensively. Cadmium plating appears on the supercharger oil seal rings and most of the propeller shaft in addition to the more common points such as valve springs, valve rockers, push rods, and impeller shaft. Chromium plate is used on the under side of the inlet valve head and on upper piston compression ring outside diameters. Lead is used in the master-rod bearing bore.

A minor design feature almost universally used is threaded pins to locate bushings. The bushing and part in which it is installed are tapped after assembly, the pin screwed into place and then machined flush inside and out. This is even found in the piston pin eye of the connecting rods. The resulting sharp corners would, of course, worry us greatly.

Cylinders are numbered by banks in the direction of engine rotation. Thus, number 1F is at the bottom of the front bank between 4R and 5R, and number 1R is at the top of the rear bank.

■ Design Details

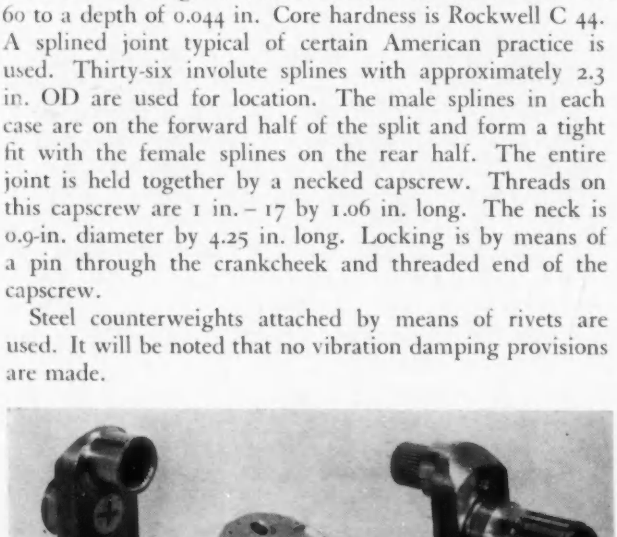
Crankcase – The crankcase (Fig. 8) is a typical three-section 17S aluminum-alloy case split on the centerline of the cylinder banks and held together by means of one

0.475-in. diameter through bolt between each cylinder. Cylinder decks are approximately 0.88 in. thick at the bore and incorporate 12 equally spaced studs for cylinder attachment. These studs are approximately 3/8-20 at the carbon, and less than 0.05% each of molybdenum, sulfur, nut end, 7/16-17 in the crankcase and have a 0.36-in. diameter neck. Cylinder-deck height is 9.8 in. approximately from the crankshaft axis. The three main crankshaft bearings fit bearing retainer rings shrunk and pinned into the crankcase diaphragm hubs. Bearing bores in the crankcase are: front, 6.65 in., center 11.13 in., and rear 6.3 in. Bearing fits at this point appear to be in accordance with conventional American practice. The front bearing retaining ring only is flanged so that the crankshaft end float is limited through the front main bearing between this flange and a steel ring attached to the aluminum-alloy cam oil transfer bracket bolted to the diaphragm. Studded to the main crankcase is a cast magnesium-alloy section which mounts the valve tappet assembly and in which a fourth main bearing is supported by a diaphragm. This intermediate casting, together with the main crankcase, forms a housing for the valve gear. See Fig. 9.

Unfortunately, the complete crankcase front section which had housed the reduction gear is not available for inspection. This nose section had been of conventional structure as shown in the front view of the engine.

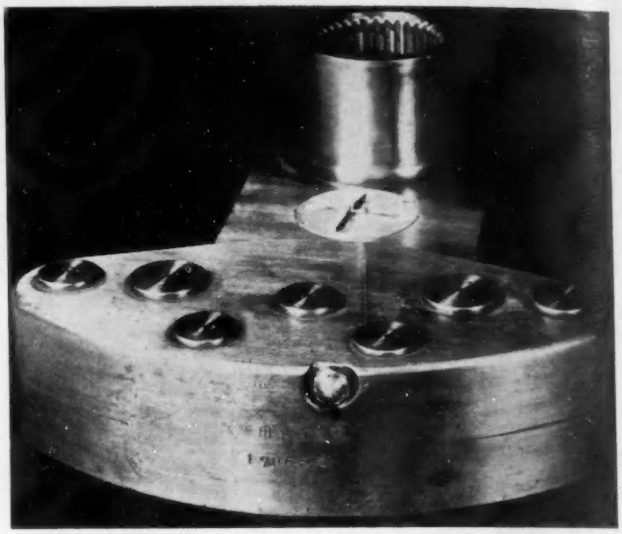
Crankshaft – The engine crankshaft is a three-piece steel shaft mounted on four main bearings as just mentioned. Fig. 10 shows crankshaft parts. Crankpins are 3 in. in diameter by 3⅜ in. between cheek faces. The installation of the one-piece master rod is accomplished by splitting the shaft near the center of the crankpin. Crankpin diameters and abutting surfaces are carburized to Rockwell C 60 to a depth of 0.044 in. Core hardness is Rockwell C 44. A splined joint typical of certain American practice is used. Thirty-six involute splines with approximately 2.3 in. OD are used for location. The male splines in each case are on the forward half of the split and form a tight fit with the female splines on the rear half. The entire joint is held together by a necked capscrew. Threads on this capscrew are 1 in. – 17 by 1.06 in. long. The neck is 0.9-in. diameter by 4.25 in. long. Locking is by means of a pin through the crankcheek and threaded end of the capscrew.

Steel counterweights attached by means of rivets are used. It will be noted that no vibration damping provisions are made.



■ Fig. 10 – Crankshaft parts

Rear section — Center bearing and retainer — Center section — Joint bolt — Front section



■ Fig. 11 – Crankshaft rear section showing counterweight and oil jet. These features are duplicated on the front section

Main bearings of NSK manufacture are used as follows: rear, sixteen 18x18-mm rollers, 3.54 in. ID by 6.3 in. OD by 1.14 in. wide; center, twenty-three balls 8.5 in. ID by 11.1 in. OD by 0.94 in. wide, symbol 8075GA; front, nineteen 17x17-mm rollers, 3.9 in. ID by 6.7 in. OD by 1.06 in. wide outer race and 1.18 in. wide inner race, symbol 8692HA. Inner races of both front and rear bearings are conventional two-piece construction. The fourth main bearing is the same size as the rear bearing except that the inner race is integral with the hub of the reduction driving gear. The outer race is positioned by the bearing ring flange and a steel snap ring. This bearing carries symbol 8707HA. Unfortunately, the symbol on the rear bearing was partially destroyed by fracture of the bearing race. Center main bearing mounting on the crankshaft is accomplished by means of a split T-section ring. The halves of this ring are attached from opposite directions to the crankshaft by means of seven capscrews each. The flange for this attachment is extended radially outward to form a sidewise locating flange for the bearing. A 0.156-in. thick tongue extends into the space provided by the difference in crankshaft bearing journal OD and bearing race ID; thus the radial load is carried on this lip. This method of mounting differs only in detail from that used on certain American engines for a similar application.

The front extension of the crankshaft front section incorporates, in addition to the front main bearing journal, 3.54-in. OD square splines for mounting the reduction driving gear. There are fourteen splines spaced on the basis of fifteen with one omitted. The designers apparently found it desirable to index the driving gear. The gear is retained by a large nut per conventional practice. The inside of the extension is bored out to receive a 2.12-in. ID copper-lead-lined heavy steel backed bushing for supporting the rear propeller shaft journal. The rear extension of the crankshaft rear section mounts the rear main bearing with conventional retaining nut and is splined internally to receive a coupling for connection with the starter and accessory drive shaft.

The crankshaft is drilled for lubrication of connecting-rod bearings and all parts forward. The oil passage through the center cheek is of some interest in that a large

■ Fig. 12 – Pistons and connecting rods

Pistons
Articulated rod
Master rod
Knuckle pins
Piston pin with plugs

axial hole serves as a point to start diagonal drilled oil holes to each crankpin. These oil holes are offset to allow the drilling spindle to miss the crankpin. The large hole is then plugged by means of a 17S aluminum-alloy spool pressed in and not otherwise retained. Lubrication to the master connecting-rod bearings is by means of five holes in each crankpin. Four of these holes are located (two on each side) in a plane normal to the plane of the crank throws. The fifth hole is close to the center of the pin, a few degrees in advance of the plane of the crank throws on the side unloaded by rod inertia. Oil jets for piston lubrication are provided by drilled holes through the counterweights to the main journal bores (see Fig. 11).

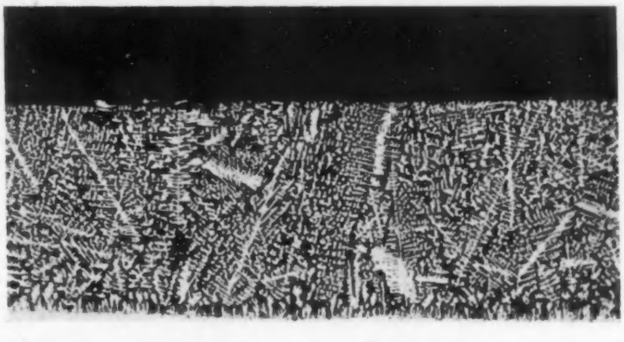
Connecting Rods – The connecting-rod system in each bank is of the conventional master articulated type. The master-rod length is 11.25 in. from crankpin to the piston-pin center. It is of I-section construction with the typical carving required for transfer from hub to shank sections. See Fig. 12. The hub section has the appearance of being rather small compared with the rest of the rod, with flanges scalloped quite closely around the knuckle-pin bores. Material is the all-purpose steel mentioned in the "General Discussion," hardened to Rockwell C 40. Master rod bearing is a heavy-steel-backed, copper-lead-lined, shrunk-in bearing with a flange at one end. Steel is soft, 0.094 in. thick. Lining analysis corresponds to American practice with a small amount of tin and 1% silver. The lining structure is good for medium loading. Bond and fracture examination were good, ductility – good, X-ray – good. Micro examination shows good distribution but coarse structure with irregular dendrites in the cross-section and shrinkage in the surface structure. See Fig. 13. Lining is 0.020 in. thick. The flange is cut away at two points to mate with keys milled into the rod hub to prevent rotation. As mentioned previously, the crankpins are 3 by 3⅜ in. The bearing shell is chamfered and cut off to provide 2.87 in. bearing length. Bearing clearance used is approximately 0.005 in.

Articulated rods are 8.7 in. long between knuckle-pin and piston-pin centers. They are the conventional I-section rods and appear very similar to some used in this country. Articulated rods are tin-bronze bushed at each end. These are very good quality castings. The material is uniform and unusually free from foreign inclusions. Hardness is Rockwell B 70. The knuckle-pin bushing is 1.03 in. in diameter by 1.57 in. long. Piston-pin bushings on both master and articulated rods are 1.25 in. in diameter by 1.81 in. long.

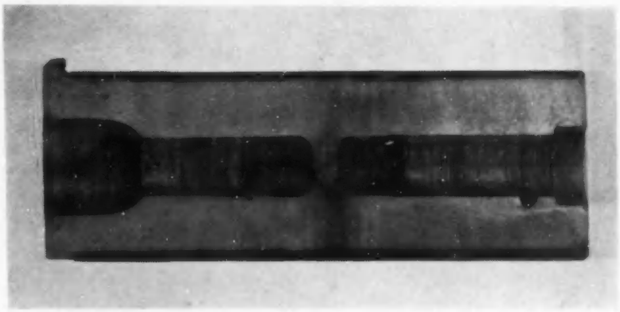
Knuckle pins (Fig. 14) are flanged at one end and locked in the master rod by lock plates screwed to the rod flange in the conventional manner. Pins are drilled from both ends leaving a web at the center. The end opposite the flange is plugged for bushing lubrication passage. As noted previously, the all-purpose steel is used. The specimen examined showed 0.040 in. case depth, Rockwell C 57 hardness on the case and 43 on the core.

Rod weights are etched on each end of each rod and are approximately 0.93 lb for the knuckle-pin end and 1.05 lb for the piston-pin end of the articulated rod; 9.13 lb for the hub end and 1.72 lb for the piston-pin end of the master rod. Equivalent rotating master weight used is 37.6 lb. Apparently no correction is made for knuckle-pin displacement or for crankpin oil. Master rods are installed in 3F and 3R cylinders.

Cylinders – Cylinder construction is of nitrided steel barrel, aluminum-alloy head type, similar to American practice. See Fig. 15. Barrel cooling fins 0.45-in. deep are machined on the steel barrel. There are 21 fins covering a longitudinal length of 2.75 in., that is, a spacing of 0.131 in. which is quite close. Attaching flanges are also turned onto the barrel and spotfaced for flat washers at the cylinder attaching nuts. A skirt length of 2.95 in. allows approximately a 2 in. projection into the crankcase interior.



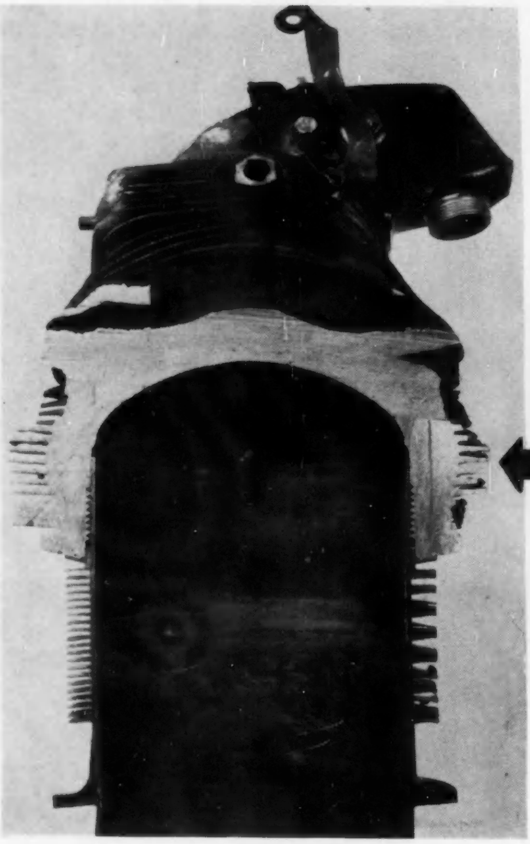
■ Fig. 13 – Cross-section of master rod bearing lining – Note coarse irregular dentritic formation. Shrinkage cracks filled with lead appear at the surface. Lining is lead plated after boring



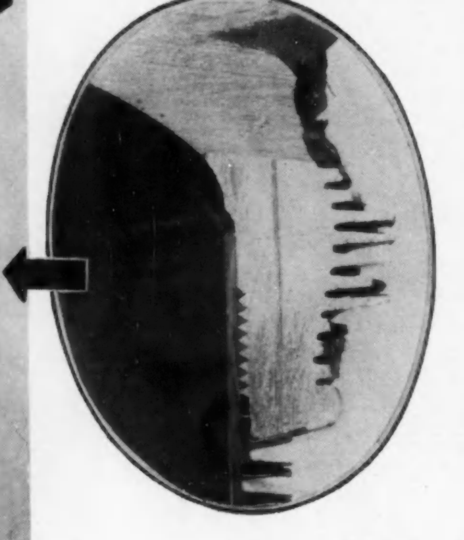
■ Fig. 14 – Section of knuckle pin showing case hardening, rather rough bores and sharp corners, and also web which supports against ovalization and confines pressure oil

Cylinder heads are characterized by quite closely spaced (5 per in.) fins which average approximately 0.9 in. in depth. This design would appear to give relatively small cooling area for the output which could be expected from an engine of this size. Relatively small angle between valves (56 deg approximately) further hinders the application of fins at top of the combustion-chamber dome.

Attachment of the head to the barrel (Fig. 16) is by means of a screw joint using threads of 3-mm pitch (8.5 threads per in.). The thread form is believed to be the International Screw Thread standard 60-deg thread with radius tips and roots. A shrink-fit pilot is provided both above and below the threaded section of the joint. The joint is completed by screwing the tapered lower face of the head against the upper side of an angularly machined fin. Presumably, the parts are machined with a differential angle so that the tip of this angular fin bears first during assembly. The cylinder-barrel threads run out into a relief. Head threads are milled and no relief is provided. It will be noted that there is a relatively long heavy barrel section within the vicinity of the angular fin which is broken up by the thread relief above and the normal fin root below. Excessive stress concentrations would be expected at the thin wall sections adjacent to the heavy section.



■ Fig. 15 – Cylinder section showing combustion-chamber shape, extreme piston-ring position, and general structure



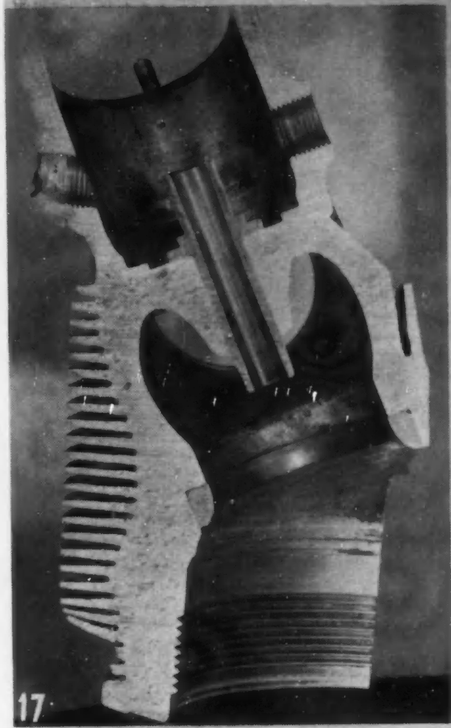
■ Fig. 16 – Method of attaching cylinder head to barrel – Note two pilot fits, thread fit, and abutment on conical fin

The cylinder head is cast of the aluminum alloy described previously. It shows a Brinell hardness number of 60.

Two spark-plug inserts are screwed into the head. The left-hand threaded joint is tapered. No other locking means is provided. The inserts, of aluminum bronze, are located at the front and rear slightly off-center, and are approximately radial to the internal dome contour. Valve-seat inserts are shrunk into the bores in the cylinder head per conventional practice. The steel exhaust insert is alloyed with nickel, chromium, and quite high manganese with a Rockwell hardness of 87 B. Intake insert is aluminum bronze. See Fig. 17. Tin-bronze valve guides are used in both intake and exhaust. Valve rocker boxes are cast integral with the head and are very similar in form to those of one American manufacturer. The box is completely enclosed except for a small cover plate over the valve end for installation of the rocker and valve clearance adjustment. Evaluation of valve ports is impossible by inspection but they appear to be well worked out. Port diameters are as follows: intake at valve end 2.24 in., at connection end 2.16 in.; exhaust at valve end 2.18 in., at connection end 2.26 in. Connection with the intake pipe is accomplished by means of a shrunk and pinned sleeve, the outer end of which is recessed inside and threaded outside to provide a packing gland type joint with the pipe. The exhaust connection in the head (see Fig. 18) is protected by a steel sleeve of the above ID approximately 0.07 in. thick by 0.94 in. long shrunk into the exhaust-port bore. Connection to the exhaust system is accomplished by means of a slip joint tube held in place by a lug and one stud. The exhaust connector used with the installation extends approximately 3 in. to a ball joint.

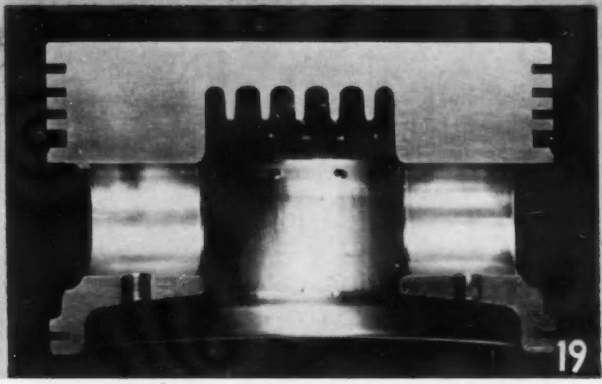
Pistons – Pistons in this engine (Fig. 19) are aluminum-alloy forgings very similar to current American practice. A Brinell hardness of 100 is quite uniform. Pin bosses are drilled for splash lubrication. Heads are flat with no valve-clearance cut-outs. The underside of the head is ribbed at right angles to the piston-pin bore. The piston is fitted with six 0.09-in. wide piston rings in five grooves. The two upper rings are flat-faced compression rings chromium plated on the outside diameter to a depth of 0.0007 in. See Fig. 20. The third ring is a tapered-face compression ring installed with the scraping edge down. There are two scalloped oil-control rings in the fourth groove. These rings are conventional in that, in addition to the scalloped lower side face, the outer face is radiused at the upper side and stepped to form oil drainage space below the scraping edge. The fifth ring, which is below the piston pin, is a typical 45-deg oil scraper. A relatively narrow land (0.23 in.) is provided above the upper compression ring. The next two lands are 0.17 and

■ Fig. 17 – Section through intake port

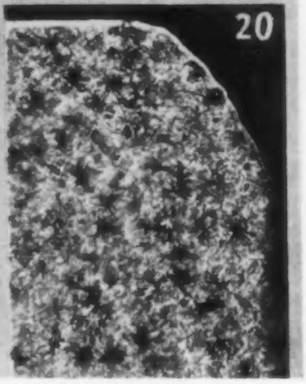


■ Fig. 18 – Section through exhaust port

■ Fig. 19 – Section of piston at pin axis



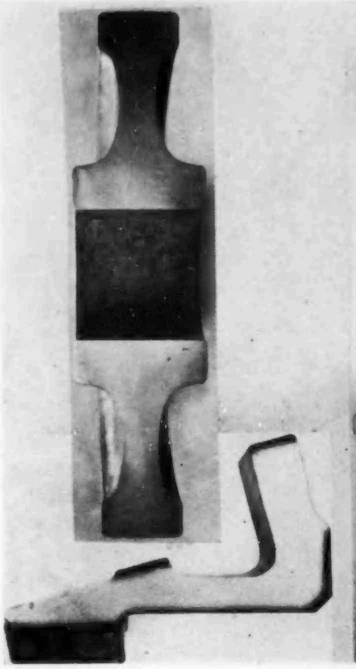
■ Fig. 20 – Section of chromium-plated compression ring used in two upper grooves. Note 0.0007 in. thick plate tapering to 0.0000 in. around 0.020 approximate radius. Note also that structure is exactly that supplied to engine manufacturers in this country

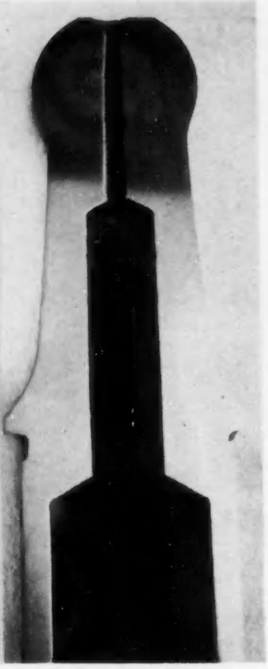


■ Fig. 21 – Section of piston-pin end

0.14 in., respectively. Ring side clearance is approximately in accordance with American practice. Scraper rings are fitted closely (0.001 in. in fifth groove, and 0.003 in. in fourth groove) with progressively increasing clearances toward the piston head (0.006 in. in the third groove and 0.008 in. in grooves one and two). All rings have parallel side faces and approximately 0.2 in. radial depth. The piston pin (Fig. 21) is a low-alloy steel hardened throughout to Rockwell C 42. It is not case-hardened. The piston pin is retained by means of 17S aluminum-alloy plugs pressed into the pin. The heads of these plugs are relatively thick and the spherical contacting area is decreased by a large chamfer. Two angular holes through this chamfer serve the dual purpose of venting the pin and providing cooling means.

Valve Gear – The cam is a double track ring running on a tin-bronze cast bushing of very good quality which, in turn, is a push fit on a ledge of the crankcase intermediate front-section diaphragm. The cam is case-hardened



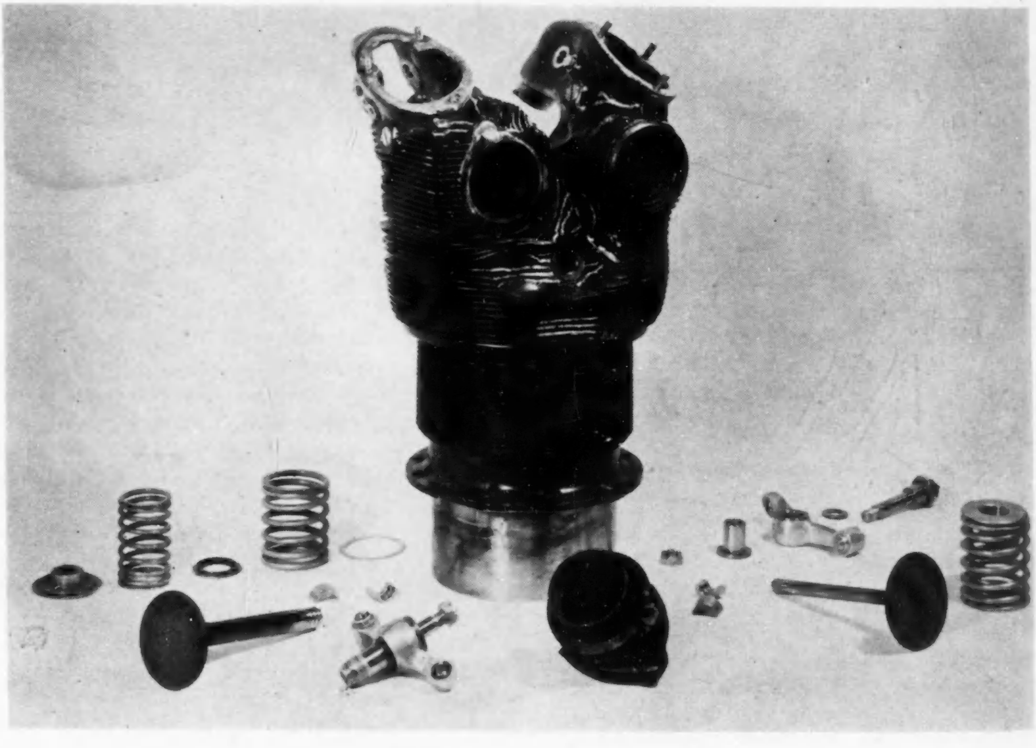


▪ Fig. 22 – Sections of cam and valve tappet roller ▪ Fig. 23 – Section of push-rod ball end

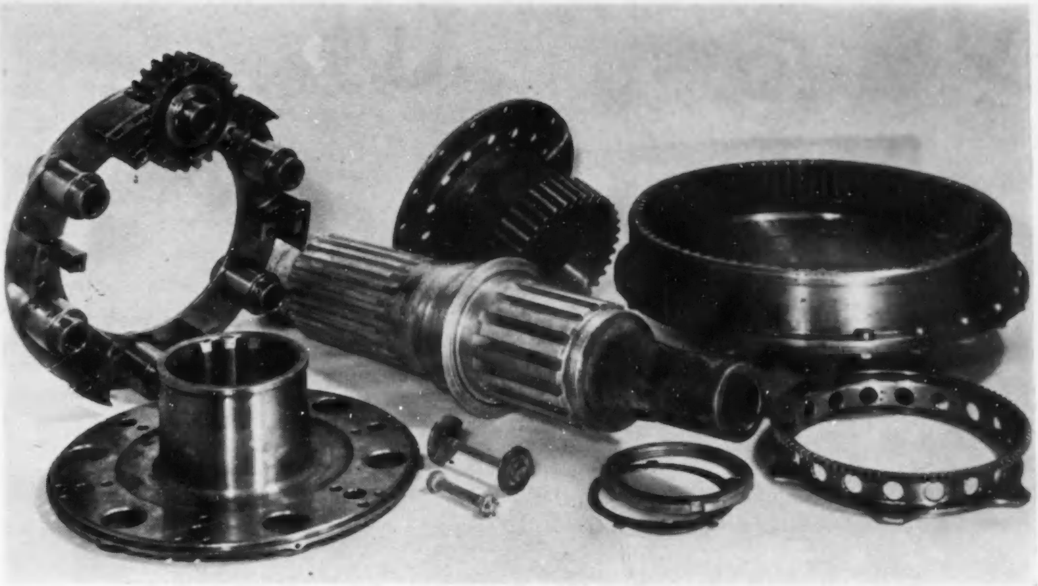
to Rockwell C 60. Core hardness is Rockwell C 32. The drive is through a pair of spur gears from the crankshaft to the intermediate cam drive. This intermediate cam drive is mounted on a stub shaft on the crankcase front main diaphragm and is made as a cluster gear incorporating a pinion which drives the internal gear integral with the cam. A bronze bushing in the cluster gear completes the assembly. It is interesting to note that no lock is provided on the screw which retains this gear, rotation being such that the right-hand thread is expected to tighten during engine operation. This gear train provides for cam rotation at one-sixth crankshaft speed and in a direction opposite the crankshaft rotation. Three lobes on each cam track provide for operation of all fourteen exhaust and all fourteen intake valves. As was noted previously, cam lobes and tappets are tilted at an angle of 14 deg-30 min to provide more nearly straight-line action of the push rods and tappets. Thrust resulting from this angle is taken through the flange of the cam bearing ring to the intermediate front section diaphragm. As a result, the designers have found it permissible to retain the cam by three short retaining pieces each held by two studs which also pass through holes in a second flange on the cam bearing ring. Clearance for the internal cam gear is provided underneath the retainers.

The cam is designed with constant-velocity pick-up and seating sectors for a running clearance of 0.045 in. $\pm$ 0.025. At 2000 rpm, pick-up and seating velocity of both intake and exhaust valves is 1.95 fps. The cam design gives 50-deg overlap, 264 deg of inlet opening, 290 deg of exhaust opening. Timing is approximately as follows although an accurate check was not made: inlet opens 20 deg early, closes 64 deg late; exhaust opens 80 deg early, closes 30 deg late. Valve lift is 0.54 in.

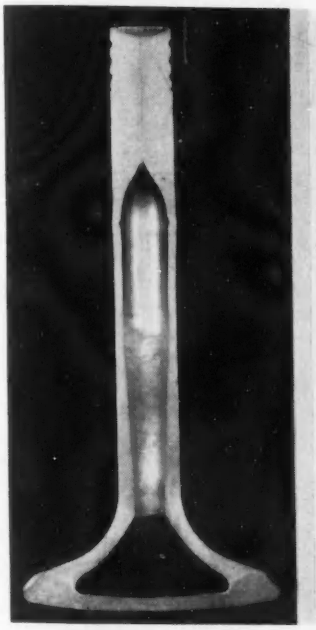
Tappets are arranged in pairs in 14 17S aluminum-alloy tappet guides (one per cylinder). A great deal of machining was done to cut these guides out of what must have been extremely simple forgings. They are bored and slotted elaborately for various reasons including oil feed and drainage. Tappets are Rockwell C 61 throughout, although the photomicrographs show a change in structure near the surface. They are 0.62-in. diameter and are fitted with pressed-in ball sockets for push-rod actuation. Tappet rollers, 1.25-in. diameter (Fig. 22), are Rockwell C 61 throughout and are mounted on 0.31 in. diameter case-



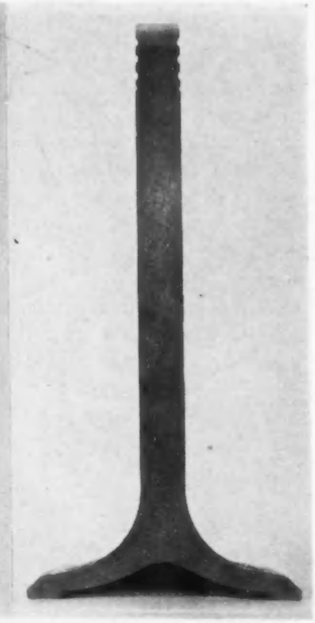
▪ Fig. 24 – Cylinder, valve-gear parts, and exhaust connection



■ Fig. 27 – Propeller reduction-gear parts



■ Fig. 25 – Section of exhaust valve



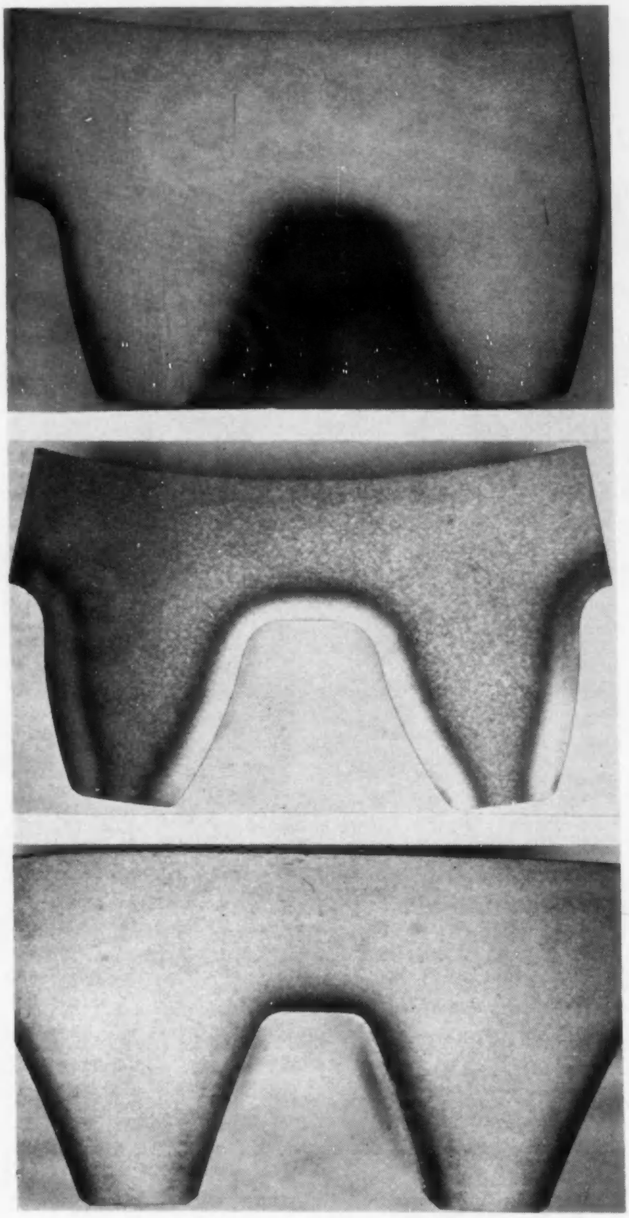
■ Fig. 26 – Section of inlet valve

hardened (Rockwell C 60 case, 30 core) floating pins. Push rods are low chrome-alloy steel tubing with pressed-in ball ends of low-alloy steel heat-treated to a hardness of Rockwell C 30, except at the tip which is quenched to obtain a hardness of Rockwell C 60. Fig. 23 shows a section of the push-rod ball end. Push-rod housings are aluminum alloy attached by means of a packing gland type joint to the cylinder rocker box. There were no lower push-rod housing connections available when the engine was inspected, but photographs of a similar engine indicate a single piece which forms attachment for two push-rod housings and is, in turn, attached to the crankcase by the three studs which also retain the tappet guide block.

Valve rockers are cadmium-plated steel forgings of the alloy described previously. See Fig. 24. They oscillate on pressure-lubricated plain tin-bronze bushings pressed and pinned into a bore in the arm. These ride on a flanged steel journal supported by a stepped rocker bearing bolt. Rocker thrust is taken by the bushing flange against a shoulder on the journal. The push-rod ball socket is permanently installed in one end of the rocker. Adjustment is at the valve end by means of a screw threaded into the arm and locked by means of a jam nut. A flatted ball bears on the valve stem and is seated in the adjusting screw, providing a familiar type of construction.

Hollow-head and -stem exhaust valves (Fig. 25) and the familiar "tulip" head solid-stem intake valves (Fig. 26) are used. The exhaust valve steel is the high-chromium, high-nickel plus tungsten and cobalt alloy generally used in this application. It is forged and machined in one piece with welded stellite tip and face. Face and tip hardness is Rockwell C 56; stem, Rockwell B 96; and head, Rockwell B 93. Metallic sodium is used as a coolant. The inlet valve is a familiar material with 13.2% tungsten, 3.2% chromium, 0.8% nickel, 0.1% cobalt, 0.4% manganese, 0.4% silicon, and 0.5% carbon. It has a hardness of Rockwell C 35 to 45 with the tip hardened to 55.

Major valve dimensions are as follows: exhaust, 2.53-in. diameter head, 45-deg face, 0.62-in. diameter stem; intake, 2.67-in. diameter head, 45-deg face, 0.43-in. diameter stem. Valves seat on inserts in the cylinder head as mentioned previously. The bronze intake insert is 2.75-in. OD by 2.24-in. ID; the steel exhaust insert is 2.67-in. OD by 2.18-in. ID. Valve-spring upper washers are retained by a split lock incorporating a tapered OD and a corrugated ID



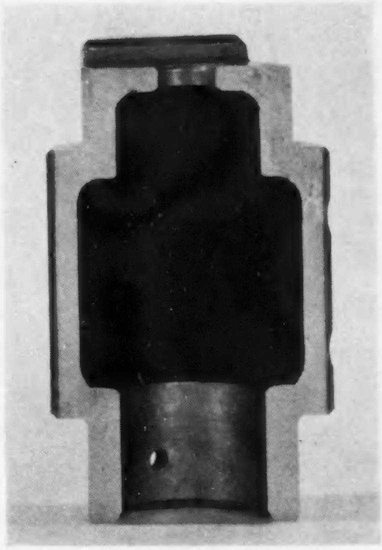
■ Fig. 28 – Sections through reduction-gear teeth – driving gear (top), pinion (center), sun gear (bottom)

which fits three circumferential semi-circular grooves in the valve stem. Two springs are used per valve – the inner seating on a washer on the guide flange and the outer on a loose steel washer in the cylinder. Springs are cadmium-plated carbon steel with a hardness of Rockwell C 40. Quality is very good.

Reduction Gear – The 0.7:1 propeller reduction gear is of the planetary type; parts are shown in Fig. 27. A large internal gear with 84 teeth is splined to the crankshaft front extension as described previously. This gear is of two-piece construction, being made up of a flange integral with the splined hub. The internal ring gear is attached to the OD of this flange by means of a large number of small diameter through bolts. The roots and flanks are Rockwell C 62. Core hardness (including tips) is C 26. The 36-tooth sun gear of this planet set is attached to the crankcase front section by through bolts in the conventional manner. Roots and flanks of this gear are Rockwell C 60. Core hardness (including tips) is C 38. (See Fig. 28.) Unfortunately, as mentioned previously, this section is not available for inspection. Six 24-tooth planet pinions are mounted on trunnions pressed into a machined-out split cage. Pinion roots and flanks are Rockwell C 59. Core hardness (including tips) is C 41. Case depth is 0.045 in. Trunnions (Fig. 29) are low-alloy steel carburized on the journal surface only to give Rockwell C 59 on the case, 42 on the core, and a case depth of 0.035 in. Pinions run on pressed-in steel-backed copper-lead lined bushings. The lining is 0.020 in. thick, of coarse structure but otherwise of very good quality and satisfactory for its purpose. The pinion cage is splined to the propeller shaft and retained in place by a large nut. The propeller shaft is the steel mentioned previously as being similar to AMS-6254. It is hardened throughout to Rockwell C 59. The propeller attachment is not common to American standards. Splines are involute type – 22 single and one wide spline cut on the basis of 24 splines. Outside diameter is 3.725 in. and spline depth 0.135 in. Cone seat diameters are 3.735 in. for the large cone, and 3.228 in. for the small. A 1-in. wide undercut is machined between the latter and the spline ends. Propeller nut threads are 2.5-mm pitch x 80-mm diameter. A small gear is bolted to the pinion cage to provide some type of drive on the crankcase front section. This gear forms the basis for the supposition mentioned under discussion of this section.

Supercharger and Drive – A gear-driven centrifugal supercharger turning at 8.48 x crankshaft speed is incorporated in the engine. The drive is accomplished in a manner very similar to that used on an American engine. The main accessory drive and starter shaft, driven through a splined coupling from the rear main bearing journal and running in a bronze bushing in the supercharger rear cover, serves a number of purposes. See Fig. 30. The hub for a spring-loaded supercharger drive gear is integral with this shaft. The impeller shaft rides on two steel-backed, copper-lead lined bushings on journals of this shaft. The shaft itself is of the material described previously having a core hardness of Rockwell C 40 and a case of 55 at wear points. The single-speed supercharger

■ Fig. 29 – Section of reduction - gear pinion trunnion



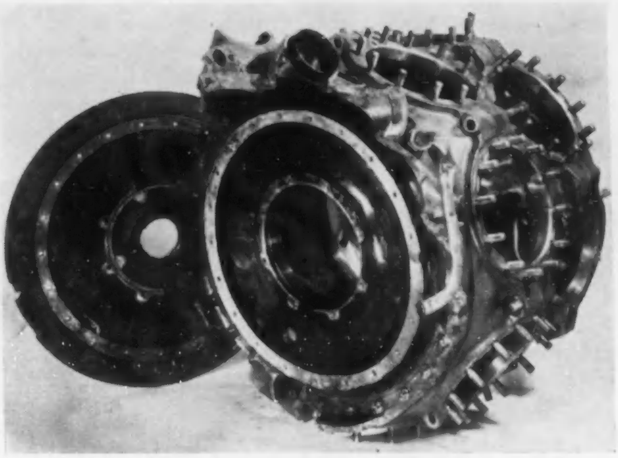
drive is completed by a case-hardened cluster gear and pinion mounted on a shaft fixed in the supercharger rear housing and piloted in a bushed bore in the supercharger rear cover. This intermediate drive cluster incorporates a copper-lead lined, steel-backed bushing. The 17S aluminum-alloy impeller is mounted on square splines on the impeller shaft just mentioned. A steel bushing is incorporated in the impeller. Impeller diameter is 9.62 in. Impeller design is conventional with 12 vanes apparently machined and bent per American practice. A 14-vane supercharger diffuser plate of magnesium alloy (Fig. 30) is mounted by means of 14 screws to a supercharger rear housing flange. Fourteen intake pipes are taken tangentially from the annulus formed between the supercharger front and rear housings, the oil baffle plate and the diffuser plate. The supercharger entrance passage from the carburetor is conventional but appears to be slightly small for an engine of this size. Axial clearance in the entrance is low.

Supercharger oil sealing is accomplished by four cadmium-plated cast-iron piston rings in impeller shaft spacer grooves at either end of the impeller. The rings seal against steel sleeves tightly fitted into the supercharger rear housing and the crankcase oil baffle plate. See Figs. 31 and 32. It is interesting to note that a boss for venting the supercharger oil seal is cast into the supercharger rear housing but left undrilled.

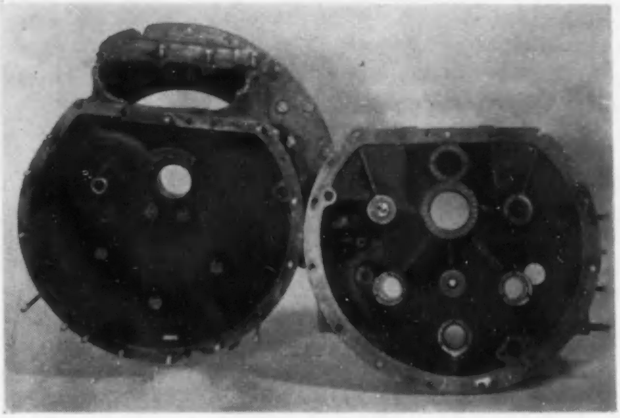
Accessory Drives – The 50-tooth spring-loaded accessory drive gear mentioned previously also drives all of the accessories except the magnetos through a centrally located 19-tooth idler gear to: (1) a 29-tooth generator drive gear and shaft; (2) a 40-tooth oil-pump drive gear and shaft; (3) a 40-tooth accessory gear box drive gear and shaft. An 8-tooth spiral gear is machined into the oil-pump drive shaft and mates with a 9-tooth spiral gear on the fuel-pump driveshaft. This forms a lateral fuel-pump drive on the left side of the engine at 1.11 engine speed. The square shaft and square pad, formerly standard on American engines, are used for the fuel-pump mounting.

Magneto drive (Fig. 33) is accomplished from a 30-tooth spur gear integral with the crankshaft extension through an intermediate magneto driveshaft which runs in two bronze bushings in the supercharger rear cover. Machined integral with this shaft are a 24-tooth spur gear and a 14-tooth bevel gear. The bevel gear mates with two 20-tooth bevel geared magneto shafts mounted laterally in bronze-bushed support housings which are, in turn, mounted in the supercharger rear cover. See Fig. 34. No oil seals are provided. Three-stud flange mounted magnetos are mounted on either side of the rear housing and are driven through a splined coupling engaging the female splines in the magneto gear shafts.

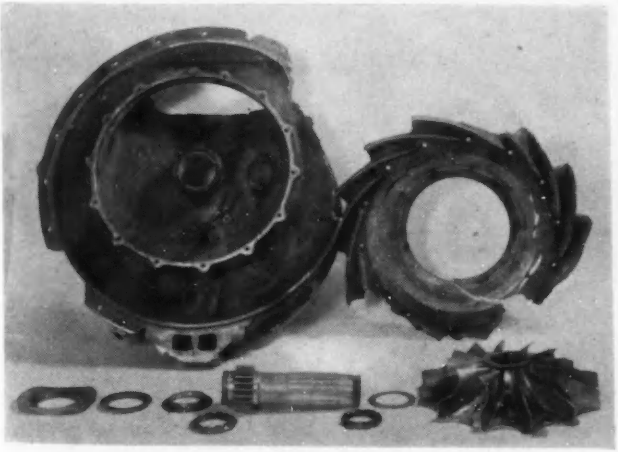
Lubrication System – A three-section oil pump comprising a pressure pump and two scavenge pumps is mounted on the rear cover. Oil from the pressure pump is taken through passages in the supercharger rear housing and a disc-type oil strainer to the large bronze bushing in which the anti-propeller end of the crankshaft extension runs. Oil transfer to the drilled crankshaft extension is accomplished through slots in the bushing and drilled holes in the shaft journal. All forward engine lubrication is taken through this journal and on through the drilled passages in the crankshaft. Master connecting-rod bearing lubrication was mentioned previously. Knuckle-pin oil is bled from the master-rod bearing clearance through holes



▪ **Fig. 31 – Oil baffle plate (left), main crankcase and remnants of supercharger front housing**



▪ **Fig. 32 – Supercharger rear housing (left) and cover – which together form the housing for the accessory drives**



▪ **Fig. 30 – Supercharger rear housing – diffuser plate – (foreground, left to right) thrust bearing and retaining parts – impeller shaft – impeller**

(Note: The two sleeves carrying four piston-type oil sealing rings each are not shown)

drilled near one end of the bearing. These holes in the shell connect holes drilled in the rod flange and thence to corresponding holes in the hollow knuckle pin. Piston-pin lubrication is by splash. Holes for this purpose are drilled in the articulated rod eye near the shank and in the bottom of each pin boss in the piston.

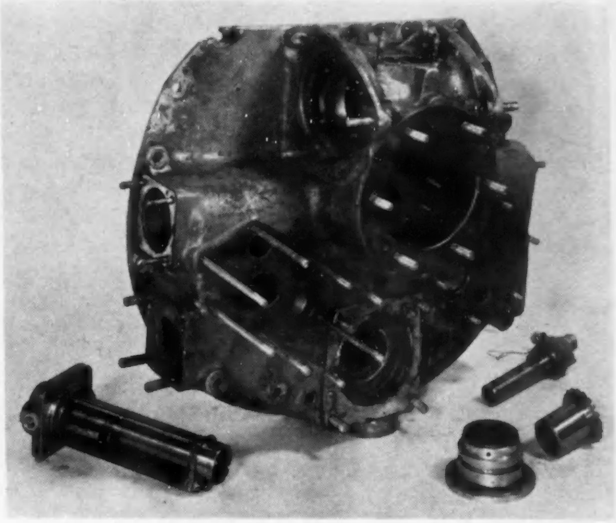
Propeller reduction-gear oil is taken from the hollow front crankshaft journal and propeller shaft through holes in the splined mount for the pinion cage and on through drilled passages to the hollow pinion trunnions.

Valve-gear lubrication is through a ring-sealed sleeve and a spring-loaded tube to the intermediate cam drive gear bracket, thence through a slip joint to the crankcase front intermediate section diaphragm and drilled passages therein to the cam ring and valve tappets. See Fig. 35. Pressure oil is metered to all plain rocker bearings through passages in the tappets, push rods, and valve rockers.

Accessory drives are lubricated through drilled passages in the supercharger rear cover leading from the main oil annulus around the crankshaft extension bushing.

This source also supplies a two-position propeller control valve in the right side of the rear cover. Oil from this valve is led through drilled passages in the supercharger housings and crankcase front sections and through tubes in the crankcase main sections to a journal-type seal on the propeller shaft. This seal is mounted within the stationary reduction gear. A spun tube in this gear completes the two-position hydraulic propeller control system. A diaphragm in the propeller shaft separates propeller oil in the forward part from engine oil in the after part.

Scavenging of the main portion of the engine is accomplished by drainage to an oil sump mounted at the bottom of the supercharger front housing. The main scavenge section of the oil pump draws oil from this sump and discharges it to the external system in the conventional manner. The third section of the oil pump takes rocker box scavenge oil from a small oil sump mounted on No. 1 front cylinder head at the extreme bottom of the engine. Oil from the rocker boxes on Cylinders 1, 2, 3, 6, and 7 front and 3, 4, 5, and 6 rear drains from box to box into this sump. Upper cylinder boxes drain through push-rod housings and tappet guides directly into the valve-gear drive compartment.



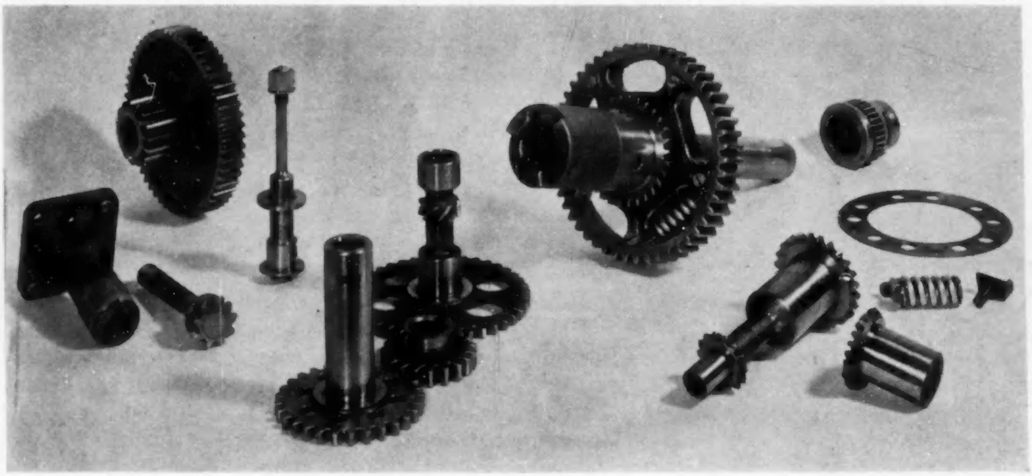
■ Fig. 34 – Supercharger rear cover mounting

Breather
Magneto
Magneto
Starter
Oil strainer
Propeller control valve
Oil Pump
Fuel pump
Accessory drive box
Generator
Detached Parts:
Oil strainer
Propeller control valve
Magneto shaft mounting and shaft

Accessories and Miscellaneous –

Information on accessories for this engine is very meager.

An electric inertia starter is mounted on the conven-



■ Fig. 33

Intermediate supercharger drive gear
Idler shaft
Fuel-pump drive support and shaft
Oil-pump drive gear
Idler with bushing partially removed
Generator drive gear
Accessory drive and starter shaft
Crankshaft coupling
Accessory gear retainer
Spring and buttons
Magneto driveshaft
Magneto drive gear

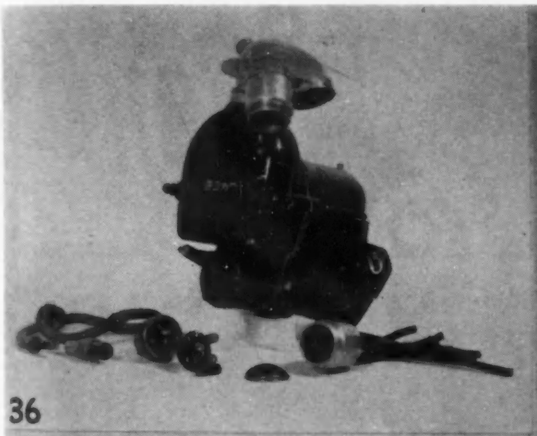
■ Fig. 35 – Oil-transfer parts supplying valve gear from crankshaft supply

tional six-bolt starter pad and engages a three-jaw end of the crankshaft extension.

The magneto photographs, Figs. 36 and 37, presented herewith are believed to be from a magneto similar but not that used with this engine. The remainder of the ignition system is radio-shielded in a manner very similar to American engines, including spark-plug elbows and spring contactors in the spark-plug well. Illustrated with the complete magneto is an interesting quick-disconnect fitting which makes it possible to remove the radio shielding from the magneto without disturbing the blocks and wire attachment. Seven wires pass through each of the blocks; however, as mentioned before, this equipment is not from the engine on which this report is based, but represents one system which is in use.

No carburetion data on this engine are available to the writer.

An accessory drive-gear box, Fig. 38, is mounted on the right-hand side of the rear cover. This box forms the drive for a single tachometer and two accessories, the nature of which is not known. This drive involves a small spur gear (probably 12 teeth), which is not available, splined into the right-hand accessory drive-gear shaft. Fig. 39 shows drive parts. It meshes with a 30-tooth gear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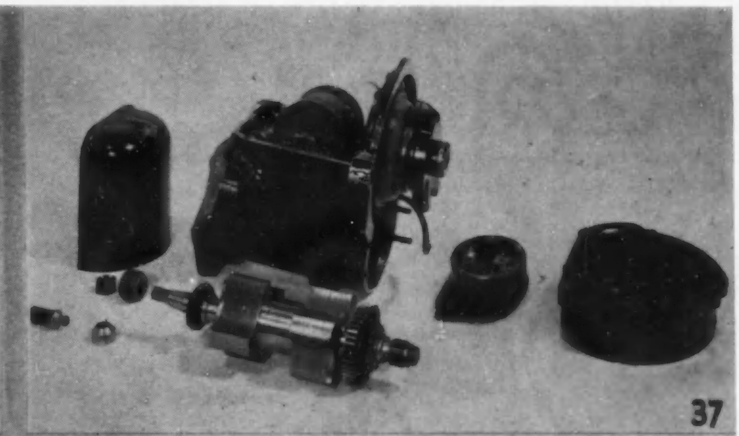
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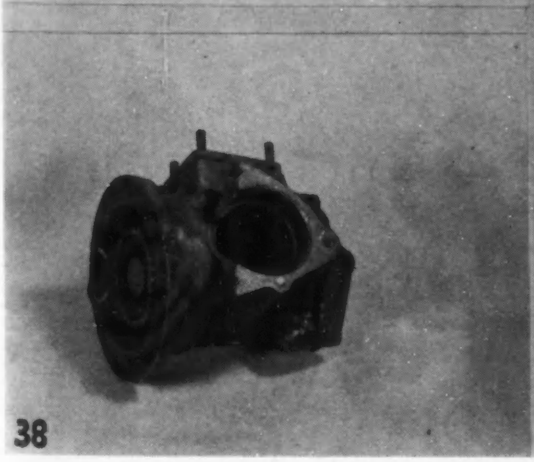
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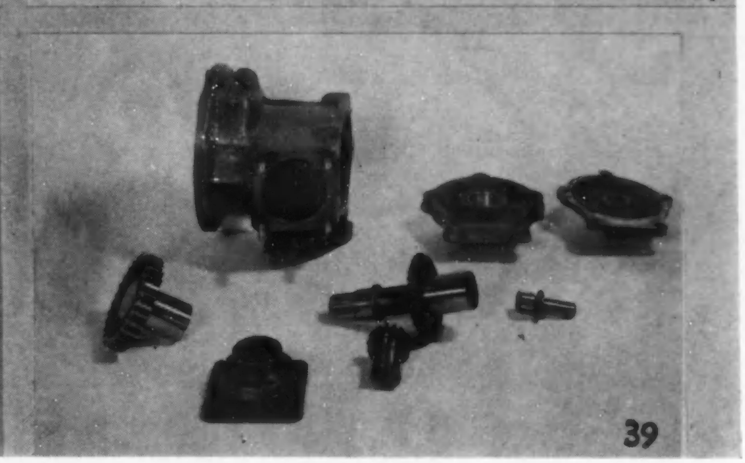
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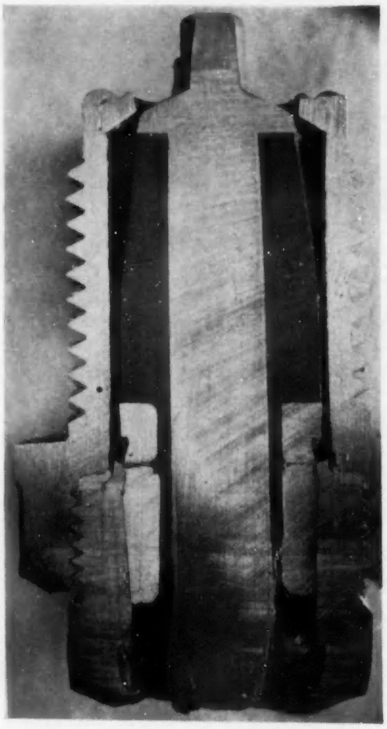
■ Fig. 36 – Japanese 14-cyl magneto assembled – parts (left to right): spark-plug lead, portion of spark plug, and high-tension junction block parts

■ Fig. 37 – Japanese 14-cyl magneto parts

■ Fig. 38 – Accessory and tachometer drive assembly

■ Fig. 39 – Accessory and tachometer drive parts



▪ Fig. 41 – Section of cylinder end of spark plug

mounted in the accessory drive housing, and splined to a shaft mounting a bevel gear. Tachometer drive is through a splined coupling directly from this shaft. It is 0.5 crankshaft speed (if the missing gear mentioned above is 12-tooth). A square-pad drive similar to the air-pump drive on American engines and a triangular-pad drive are accomplished through two bevel gears each mating with the gear on the main shaft.

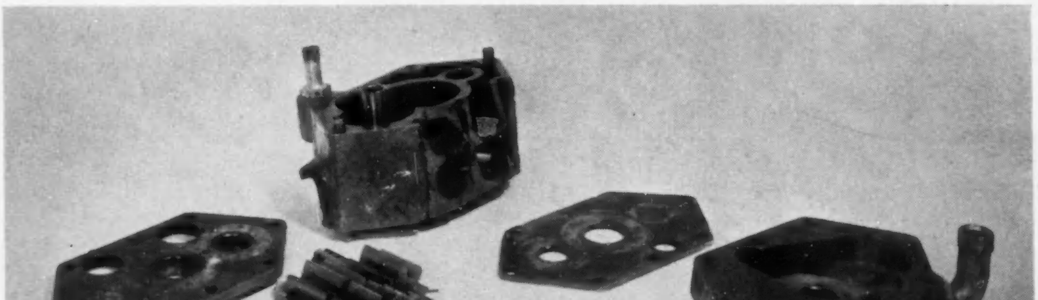
The nature of the drive used on the crankcase front section is not known, but it is believed that a combination gun synchronizing impulse generator and constant-speed propeller governor drive is made available at this point on later engines.

The oil pump is mounted at the left-rear of the engine, taking its drive through a spline in the oil pump and fuel-pump drive shaft mentioned previously. Parts are shown in Fig. 40. A magnesium-alloy housing is cored for oil passages and mounts the gears and shafts directly. The 1.12 in. wide 9-tooth scavenge gear is splined to the engine shaft and drives the main oil pump shaft on which three 11-tooth gears are mounted. The 0.88 in. wide oil pressure pump consists of a gear keyed to this main shaft driving a 9-tooth idler. Both of these pumps are in the main pump housing. A thin plate separates the pressure pump from the valve gear scavenge pump which is a duplication of the pressure pump except that the teeth are only 0.47 in. wide. The main oil pump shaft is also fitted with a floating member to provide a tongue drive for a square pad accessory on the rear of the valve gear scavenge pump housing. A quill drive for this accessory is formed by a 0.27-in. diameter by 3-in. long neck between the splines which fit into the forward end of the main shaft and the slotted journal. All oil-pump gears are carburized low-alloy steel.

The remains of a spark plug is sectioned and shown in Fig. 41. It is a mica-insulated plug of quite conventional construction.

Engine mounting is accomplished by means of seven longitudinal bolts in bosses cast at alternate intake pipe connections in the supercharger front housing.

Breathing appears to be through a flange at the top of the magneto drive shaft housing in the supercharger rear cover.



▪ Fig. 40 – Oil-pump parts

Portion of main oil-pump body
Front end plate
Central body plate
Gears and shaft
Valve-gear scavenge body

FUELS and LUBRICANTS

procured by The Quartermaster-General

for U. S. Army Motorized Ground Forces

by GEORGE A. ROUND

Consultant on Fuels and Lubricants,
Motor Transport Service, The Quartermaster Corps

PRACTICALLY every selection of fuels and lubricants for the Army Motorized Ground Forces must be a compromise among three considerations – the requirements of the units, the availability of the desired products, and the problems of supply, Mr. Round points out.

Stressing the need for simplification of the supply problem, he brings out that, whereas in civilian operations, a mistake may be merely an inconvenience; in battle it may cause a major disaster. He shows that such simplification is not a simple problem by reporting that practically every type of engine is used in the Ground Forces – gasoline, diesel; air and water cooled; in-line, vee, and radial designs, ranging in size from tiny lighting units to the engines used in the largest tanks. The Army problem, he summarizes, is to reduce the wide variety of fuels and lubricants recommended for these engines and vehicles in civilian service to the absolute minimum consistent with satisfactory results. The major part of his paper outlines the real progress that has been made toward this objective.

The seven Appendixes of the paper give the pertinent condensed U. S. Army specifications.

★

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THREE primary considerations dictate the selection of fuels and lubricants for Army Motorized Equipment: first, the requirements of the units; second, the availability of the desired products; and third, the problems of supply. It is difficult to say which of these is the more important; there are continual conflicts between them, and practically every selection has been a compromise.

■ Importance of a Simple Program

One can never forget the problem of supply at the front. The more simple it can be made, the more certain we can be that every vehicle will be serviced properly. The ideal set-up would be one fuel for all engines – one lubricating oil that would take care of both engines and gears – one grease for all chassis lubrication. If oil could be substituted for the grease, so much the better.

It isn't a military secret that diesel fuel has been put into gasoline tanks, and vice-versa, or that gear oil has gotten into engines. Such things have happened in civilian motor transport but, in the excitement and pressure of battle conditions, the chances for grabbing the wrong can or drawing from the wrong drum are infinitely greater. Hence, the more the supply problem can be simplified, the more certain we can be that the right product will be used. In civilian operations a mistake may be merely an inconvenience; in battle it may prove to be the cause of a major disaster. That possibility should always be dominant in the mind of anyone who has anything to do with the war effort today. In the words of a well-known petroleum member of the SAE now serving overseas: "We are playing for mighty high stakes – our necks are on the table."

Unfortunately, it is not possible to simplify the supply problem to reach the ideal just described; the demand for an enormous number of units in the shortest possible time dictates that every available design, suitable for military use and capable of immediate mass production, must be utilized. In the Ground Forces for which the Office of The Quartermaster-General procures fuels and lubricants practically every type of engine is used. There are both gasoline and diesel types, air and water cooled, with in-line, vee and radial designs ranging in size from tiny single-cylinder air-cooled lighting units to the engines used in the largest tanks. Both babbitt and copper-lead bearings are employed, the latter being used in several units including the 2½-ton truck which is the most numerous

[This paper was scheduled for presentation at the 1942 Semi-Annual Meeting of the Society, which was ruled out because of transportation priorities.]

of Motor Transport vehicles. Since these designs differ in their fuel and lubricant requirements, the manufacturers have recommended a wide variety of products for them in civilian service. The Army problem, therefore, has been to reduce these to the absolute minimum consistent with satisfactory results. Some real progress has been made along that line as will be outlined in the following paragraphs.

Fuels

As mentioned previously, both carburetor and diesel-type engines are used by the Army Ground Forces. While by far the larger percentage of units burn gasoline, there is a substantial number of diesels used, both in combat vehicles and in auxiliary equipment, such as tractors. Hence, both types of fuel must be procured by the Quartermaster-General.

From the supply standpoint, it is mighty essential, if not imperative, that all Army ground equipment operate satisfactorily on commercially available fuels. In other words, any tactical unit comprising combat and transport units with auxiliary equipment should be able to use fuel from filling stations, bulk plants, or refineries along any route that it travels. It should not be necessary for any such units to depend upon special products laid down along the line of march or in any particular field of operation in the U. S. A. In overseas operations the use of a single fuel is even more desirable.

Gasoline Requirements

As a national economic measure, it has been necessary to conserve lead. This restriction has resulted in a general lowering of the octane level of gasoline, both premium and regular, throughout the country. In the case of certain refiners, this trend has also made possible the release of stocks which could be utilized in making aviation gasoline. However, this downward trend could not continue indefinitely without coming into conflict with Army requirements.

A substantial number of combat vehicles demand gasoline of relatively high octane value. While much work has been done to keep their requirements within the range of premium gasolines, as normally distributed throughout the United States and readily available for export, it is a fact that there is a definite minimum limit below which it is impossible to go without seriously impairing the performance of vitally important combat units. This limit, therefore, fixes the minimum octane number requirement, not only for Army gasoline but also for premium gasoline in general distribution.

Motor transport vehicles and auxiliary equipment appear to operate reasonably well on fuels comparable with regular grades of gasoline that have been generally distributed throughout the country. However, it is obvious that the lowering of the octane level of this type of fuel could not continue indefinitely without impairing the performance of these units. Here again, therefore, we have Army needs tending to fix the octane level of commercial fuels.

While all Army transport units would operate satisfactorily on premium gasoline of suitable octane value, from the standpoint of national economy it is desirable to conserve as much premium fuel as possible. Hence, it is the present policy to use regular-grade gasoline at all bases in the United States where transport vehicles alone are operating. Where combat and transport vehicles are operating together outside of defense or combat zones, both regular and premium-grade fuels will be furnished providing there are storage facilities for both grades. In combat or defense areas where both types of units are operating, and for overseas operations only, premium-grade fuel will be used. This procedure eliminates the possibility of using the wrong grade of gasoline under conditions where it might be disastrous.

Gasoline Specifications

It is obvious from what has been said that Army gasoline specifications should be written around those of average commercial gasolines as distributed seasonably throughout the United States. This has been done, and the two grades have been designated respectively Motor Fuel (all-purpose) U. S. Army Specification No. 2-103A and Motor Fuel – 72-octane U. S. Army Specification No. 2-114.

The essential items of these two specifications are given in Appendix 1, the data shown in sections E-1-a, E-1-b, E-2, and E-3 being identical for both specifications. The following comments on the requirements may be of interest:

The distillation requirements of corresponding seasonal classifications are practically identical and may be made the same. Present differences reflect only those recently found in the field.

Regarding sulfur requirements, it will be noted that more tolerance is permitted in the 72-octane fuel. At present in certain areas, there are considerable quantities of fuel being marketed which show a sulfur content between 0.10% and 0.25%. No trouble from the higher sulfur content is to be expected in warm-weather service. In cold-weather operation, however, where vehicles are used in intermittent service, corrosion could take place with high-sulfur fuels if much condensation occurred in unventilated crankcases. It is, therefore, considered desirable to hold to a lower limit on sulfur during cold weather.

In the case of the all-purpose 80-octane gasoline, which might be used or shipped anywhere, anytime, it was felt that a low minimum sulfur content should be specified.

Item E-1-b applies to the so-called lubricated gasolines marketed by some refiners which contain a small amount of light lubricating oil. Obviously, the gum determination must be made on the fuel alone, as otherwise a fictitious value for gum content would be obtained.

The territorial layout for the use of the various grades follows almost exactly the ASTM recommendations. Those differences which do appear were made to give a better tie-in with commercial distribution as shown in the Bureau of Mines survey made last fall.

Non-Leaded Fuel

Some Army equipment, such as field cook stoves and blowtorches, also some very small air-cooled engines require, or at least operate with less service trouble on non-leaded gasoline. It is interesting to note that, instead of stocking non-leaded fuel, which is not readily available in many localities, a small filtering unit has been designed which removes the lead from leaded gasoline. This unit utilizes activated clay. After a fixed quantity of clear fuel

is recovered, the charge of clay is thrown away and a new one put in, so a simple unit serves to provide the small quantity of this type of fuel required.

Octane and Vapor-Lock Requirements

It has already been indicated that the octane-number requirements of combat units have been determined within close limits, thereby defining this characteristic of the all-purpose gasoline. Similar investigations on motor transport vehicles are now under way under the supervision of a subcommittee appointed by the CRC (Cooperative Research Council) War Advisory Committee.

Preliminary studies of these same vehicles indicate that their vapor-lock characteristics differ considerably from those of corresponding civilian vehicles. Hence, a study of these characteristics is being made under the supervision of the same subcommittee. These investigations are being carried on in connection with the Quartermaster Desert Test Command operating in the Southwest.

These and other studies may result in some revisions of the gasoline specifications now in effect as well as minor modification in the units.

Diesel Fuel

The fuel requirements of diesel engines used by the Army Ground Forces do not differ appreciably from those of similar design in civilian service. Hence, the specifications covering Army diesel fuel oil closely approximate those of commercial high-speed diesel fuels. The essential requirements for Oil, Fuel, Diesel for High-Speed, Automotive-Type Diesel Engines, the title for U. S. Army Specification No. 2-102B, are given in Appendix 2.

It will be noted that three different volatility grades are called for to cover different climatic conditions in the simplest possible way. The specifications are also so drawn that Grade B may be made by blending Grades A and X in equal proportions.

Some may question the low pour point for grade A. However, this fuel is designed for all-year use in regions where there may be sudden sharp changes in temperature. Stocks of higher pour point material might be on hand at a time when temperature changes would make them temporarily unusable so, to assure unfailing performances, the relatively low pour figure is stipulated.

Regarding cloud point, experience has indicated that, where there is a relatively wide spread between pour and cloud test figures, at temperatures slightly above the pour test, sufficient wax may separate from the fuel to cause fuel filter clogging and tie up the units. As an assurance of unfailing operation at temperatures close to the pour point of the fuel, a narrow spread between cloud and pour test limits has been specified.

Speaking of the fuel set-up in general, we believe that it is as simple as design conditions will permit, and that, from the supply and economic angles, it is sound and workable.

Lubricants – Engine Oils

Coming now to the matter of engine oils, after considering the various types of engines used, it was obvious that, if civilian experience were to be accepted as a guide, high-speed diesels should be lubricated with very stable, detergent type oils – in other words, the heavy-duty motor oils that have been developed during the past few years. The same was true for certain high-output carburetor type engines. Recent experiences, both civilian and military, also indicated that any engine equipped with hard alloy bearings should be lubricated with very stable motor oils.

Some argued that all military engines should be redesigned to use non-corrodible bearings; that no engines requiring "special" oils should be employed. But they ignored two things: first, to get high output with minimum weight, high-performance engines must be used. Bearing loads in some of these engines are beyond the safe limit for babbitt. Second, with the time limits set for the announced production goals, there is no time to redesign, and every available engine that is suitable for the service must be used.

If the supply problem were to be simplified, it was obvious that the only engine oil to be used would be that suitable for the high-speed diesel – in other words, a "heavy-duty oil." There was abundant civilian experience to indicate that this type of oil could be employed to advantage in any type of ground vehicle engine. Not only was it used extensively for commercial vehicles but, had the war not interfered, it is probable that all the major oil companies would have been marketing oils of that type through filling stations since at least two had been doing so before Pearl Harbor. Hence, with all requirements considered, a tentative decision to use only heavy-duty oils was reached last summer.

Specifications

Supplies for the Army are customarily bought on the basis of specifications. It was realized early, however, that something better than the conventional specifications for lubricants would be required if the Army was to get what was wanted. Officers of the Ordnance Department and Quartermasters Corps, therefore, invited representatives of the automotive and petroleum industries to a joint meeting in Detroit, Mich., early in September, 1941, to consider what sort of specifications should be set up.

At that meeting, it was agreed that, to be acceptable for Army Ground Force use, an oil should pass the engine tests required for a Caterpillar Tractor Co. certificate of approval and, in addition, successfully complete the General Motors Diesel 500-Hr Test for approval for field test. The latter was considered to be proof of adequate stability and the former of satisfactory ability to keep pistons and rings free from gum and hard carbon – that is, detergent properties.

Some simple physical test requirements were also set up, defining the general characteristics of the desired oils. The most important point to note in connection with these requirements is that only three grades were called for – SAE 10, 30, and 50. Agreement on this point had been reached some time before as another means of simplifying the problem of supply.

Subsequently, agreement was secured from all manufacturers furnishing engines for Ground Force units that these oils would satisfy the requirements of their designs.

Recently, these specifications were revised in some minor details on the advice of the Cooperative Research Council War Advisory Committee (formerly the CFR-SAE Fuels and Lubricants Advisory Committees) and have been adopted under the designation of Army Specification

2-104A dated April 9, 1942. In condensed form they are given in Appendix 3 of this paper.

Reference to these specifications will show that, while the SAE 30 grade must pass both Caterpillar and General Motors Diesel Tests for approval, the SAE 10 grade is required to pass only the Caterpillar Type Tests. The reason for this difference is that no procedure has been set up for testing this light grade in the General Motors diesel. However, it is anticipated that a Chevrolet 36-Hr Heavy-Duty Test will be required for this grade in addition to the Caterpillar Test, as soon as the method has been standardized by the ASTM.

It will also be noted that no engine tests are stipulated for the SAE 50 grade, the reason being that it is too heavy for satisfactory use in either the Caterpillar or General Motors diesel engines. However, it is required that the same additive be used in this grade as in the approved SAE 30 oil. Investigations are being made under the direction of the CRC War Advisory Committee to determine what engine tests can be used as a basis for approval of the SAE 50 grade.

■ Availability of Heavy-Duty Oils

When Army Specification 2-104 was first adopted, there were only two brands which met the requirements. Some of the ingredients used in oils of this general type were based on critical materials. These facts caused many to doubt whether it would be possible for the Army to use heavy-duty oils exclusively. However, during recent months, at least four more additives have been developed, and today seven basic brands of heavy-duty oil are approved. The additives used in these oils are available in sufficient quantity so that other refiners may use them in making heavy-duty oils. With present and potential production, the supply appears to be far in excess of any demands of the Army. We can, therefore, be assured that, as far as the basic problem of supply is concerned, the Army will have available for use the best oils the petroleum industry knows how to make.

■ Changeover Problems

There has been some question regarding the compatibility of the various heavy-duty oils. However, engine tests conducted with mixtures of the available, approved oils indicate that there is no difficulty in this respect, the results being about what might be expected from averaging the characteristics of the individual oils in the mixture.

Some difficulties have been anticipated in installing detergent oils in engines showing varying degrees of crankcase contamination. However, it is significant that a large number of Army units have already been changed over without difficulty. As a protection, however, some very carefully worked out instructions for draining, flushing, and operating badly sludged units have been prepared for the field forces. Since the great majority of Army vehicles are relatively new and, as units procured in the future will use the heavy-duty oils exclusively, it is believed that there will be little if any more trouble due to the cleansing properties of these oils, than there has been in their use by passenger-car owners who bought oils of this type through filling stations.

In general, it is believed that the adoption of heavy-duty oils by the Army will be excellent insurance against those types of engine failures which are justly attributable to inferior lubricants.

■ Gear Oil Requirements

About two years ago, the Army was faced with the problem of deciding whether or not to accept trucks equipped with hypoid gears. On the assurance of the motor manufacturers that there were several adequate sources of supply of lubricants suitable for service in truck hypoid axles, this gear design was accepted.

The gear lubrication problem then involved spiral-bevel, and hypoid axles, transmissions in wide variety, transfer cases for four or six-wheel drives, and also steering gears and winches. Some of these could be lubricated with straight mineral oils, some required at least a mild EP material, while the hypoid axles demanded a truck-type hypoid lubricant.

Considering the certainty of trouble if either a straight mineral or mild EP lubricant were used with hypoid gears, and also the need for simplifying the supply problem, it was decided that, with one exception, the mild, truck type, hypoid lubricant would be used for all gears. For transmission, differentials, and final drives of tanks the same type of oil as used in the engine would be employed.

■ Specifications

As a basis for procurement a Federal Specification VV-L-761, dated Oct. 1, 1940, was set up which called for three viscosity grades – SAE 80, 90, and 140. This specification stipulated the use of very stable mineral oils, meeting certain Navy Specifications and having marked ability to separate from water, to be blended with approved additives. In addition, it was required that the lubricants pass the Chevrolet "Bump" test for lead-carrying ability.

As a further step in simplifying the supply problem for areas where units might be subject to a very wide range in temperatures, this specification was modified by the adoption, in place of the original SAE 140 grade, of a special oil meeting the viscosity requirements of both the SAE 80 and 140 gear oil classification. The specifications for the three grades, with the foregoing modifications as now in effect, are given in condensed form in Appendix 4.

■ Present Gear Lubrication Problems

The problems connected with gear lubrication have not been solved completely by the adoption of these specifications. From the supply standpoint, it would be desirable to reduce the number of grades. Unfortunately, only one refiner is in a position to furnish the 80-140 grade and the supply of that is limited, so present indications are that both 80 and 90 grades at least are required.

The major problem is the transfer case. Oil temperatures in this part vary with the different designs and with operating conditions. They have been reported to exceed 275 F, which causes rapid deterioration of any oil and is particularly bad for extreme-pressure lubricants since these, to function as such, must be chemically active at high temperatures.

Leakage is also a factor which can be reduced to some extent by the use of the SAE 140 grade. On the other hand, as might be expected, the higher-viscosity oils cause higher operating temperatures which accelerate deterioration of the lubricant. They also foam more, which aggravates leakage.

The question of whether it is better to accept some increase in leakage with the SAE 90 grade and thereby

reduce temperatures and depreciation of the lubricant, or to use the heavier grade in hot weather and drain more frequently, is one which is being investigated in connection with the tests now being conducted in the Southwest by the Quartermaster Desert Test Command. Admittedly, the answer will be a compromise about which there can always be debate, but military requirements dictate that the best practical solution be found.

One other problem is under investigation, namely, the possibility for rusting when water gets into gear housings. It has always been known that, when gear lubricants containing chlorine are used, rusting is likely to take place in the presence of water unless the lubricant has emulsifying properties. From a practical standpoint this condition has not been troublesome in either civilian or military operations for two reasons: First, the manufacturers almost invariably have used for the initial fill gear lubricants with excellent emulsifying properties; second, almost any lubricant develops these properties after a short period of use. However, where new or freshly cleaned parts are lubricated with new non-emulsifying, chlorinated lubricant containing water, there is a possibility of rusting. It must be stated further that rusting also has occurred with straight mineral oils when large quantities of water leaked into the gear housings.

While experience indicates that rusting is an uncommon complaint, it is a fact that Federal Specification VV-L-761 calls for lubricants which will not emulsify. One reason for this is to get extremely stable mineral oils that will stand the high temperatures encountered in the transfer case. Another is to reduce foaming which aggravates leakage. If the possibility for rusting could be eliminated without sacrificing stability or increasing foaming, it would be a desirable improvement. This problem has, therefore, been referred to the CRC War Advisory Committee for investigation. It is hoped that, before summer is far advanced, these questions can be answered. Meanwhile, specification VV-L-761 continues in effect.

■ Chassis Lubrication

Greases for chassis lubrication generally are procured under Federal Specifications issued by the Procurement Division of the Treasury Department and published in the "General Schedule of Supplies, Lubricating and Other Oils and Greases, Class 14." The essential items in these specifications are given in Appendix 5.

A very brief study of these specifications will show that they are very broad and that practically any type of grease may be supplied in any given consistency range. There are nine grades, but some of these grades differ only in minor details as regards oil viscosity and soap content. There is no specification which would assure getting a wheel bearing grease designed for heavy-duty service, such as called for by the Timken Roller Bearing Co., and the one type which covers water-pump greases contains a requirement that shuts out practically every grease suitable for that service. There are also greases required for Army use which cannot be procured under these specifications.

At the same time, it must be admitted that it is an exceedingly difficult matter to define completely the desired performance characteristics of a grease by any standard methods of test. If a highly restrictive specification, based on chemical analysis, is set up, it may be found that a grease made to meet the specification will not be satisfactory in service, whereas another not meeting it will give superior results.

■ The Army Problem

Motorized equipment needs as good or better chassis lubricants than required by commercial fleet operations. Army transport vehicles face conditions of dust, mud, and water to a greater extent than do civilian vehicles, and they must function efficiently at arctic or desert temperatures. While at bases there may be adequate lubricating equipment, in the field hand guns may at times have to suffice.

Again there is the need to reduce the number of grades to the absolute minimum in order to simplify the supply and equipment problems. The initial step in this direction taken by the Motor Transport Service was to recommend the use of only five of the nine types of grease listed in the Treasury Department Specifications, to cover all service conditions. Of these, Types C and D were selected as summer and winter chassis greases respectively; Type G for wheel bearing lubrication under normal service conditions; Type I for extremely high temperatures when there was wheel bearing grease leakage with Type G. Type H was recommended for water pumps.

Another recommendation formerly made was that only soda-soap greases be purchased for Types C, G, and I, and that lime-base greases be procured under Types D and H.

It has been realized that the best results could not be obtained under any such set-up. Where inferior greases were secured, there has been a tendency to go to something heavier, with the result that some parts have not been adequately lubricated. The field personnel has not realized that in chassis lubrication, particularly of track rolls, better protection is afforded by a slow, steady leakage of the lubricant which continually washes out dirt, dust, and water. There was a preference for greases that didn't leak, and the possibility for channelling was ignored. Lack of lubrication also has resulted from the use of grease that was too hard to dispense easily with a hand gun. It didn't get forced in where it was needed.

■ New Grease Specifications

This problem of grease specifications was referred to the CRC War Advisory Committee for study. A subcommittee of experienced grease chemists was organized and assigned the job of recommending methods of determining grease consistency and pumpability at low temperatures and stability at high temperatures. It was also asked to suggest that improvements might be made immediately in the present Treasury Department Specifications to aid the Army in getting better greases within the limits of available test methods.

As a result of preliminary work on this problem, new specifications for greases have been set up under Army specification numbers. Five types are called for of the same consistency ranges as in the Treasury Department Specifications but with different requirements for type of soap, viscosity of mineral oil, and so on. The new specifications are tabulated in Appendix 6.

As this paper is written, not all the details in connection with the procurement of these greases under Army Specification numbers have been worked out. However, it is hoped that, through the work of the CRC War Advisory

Committee, further improvements in the specifications will be made so that the Army will be able to procure the best available chassis lubricants. It is fully realized that the latest specifications are not as complete as they should be.

■ Product Recommendations

To assure the correct use of the various lubricants that have been described, the chart given in Appendix 7 has been prepared.

It will be noted that, under any given temperature conditions, only one pressure gun or chassis grease will be required. Those units which have water pumps with grease cups will carry a 1-lb tin of the water-pump grease. Since wheel bearings are normally lubricated at 6000-mile intervals, according to Army Regulations, it is not necessary to have a supply of wheel bearing grease on the vehicle at the front. Hence, the driver or any other personnel assigned to the job of lubricating the vehicle at the front will have to dispense only one or, at the most, two greases, a chassis grease and, where necessary, a water-pump grease.

Lubricant Distribution

To reduce in the field the possibilities for contamination of lubricants by sand and dirt of all kinds, the Army is taking a leaf from the petroleum industry's experience and is supplying engine oil to the field forces in 1-qt, 5-qt, and 5-gal cans rather than in drums. Greases and gear oils are supplied in 25-lb containers, except water-pump grease which is bought in 1-lb tins. The latter containers are designed to empty directly into the gear oil or grease dispensers from which the vehicles are either lubricated directly or the hand guns are filled. Permanent bases, of course, will continue to receive lubricants in standard drums but, at such points, there is far less chance of contamination than in combat areas.

Handling lubricants in small containers allows each vehicle to carry sufficient supplies for its immediate needs and reduces the losses which occur when individual vehicles must be serviced from full-sized drums which may be upset, contaminated, or destroyed by enemy action.

■ Summary

In general, it may be said that much progress has been made by the Army in getting the most desirable types of fuels and lubricants. While further development work must be done, we can feel assured that the possibilities for serious trouble due to product quality have been greatly reduced.

Appendix 1

MOTOR FUEL (ALL-PURPOSE)

U. S. Army Specification No. 2-103A (Condensed)

E. Detailed Requirements.

E-1 Motor fuel shall be of one type with automatic provisions for locality and climatic conditions, and shall conform to the following requirements:

	Classification A	B	C
ASTM Distillation:			
Maximum Temperature			
for 10% evaporation, F	160	145	130
for 50% evaporation, F	257	240	230
for 90% evaporation, F	356	348	340
Residue, max., %	2	2	2
Vapor Pressure, max. psi (a)	8	10	12
Octane No. ASTM, min.	80	80	80
Gum max., mg/100 ml (b)	7	7	7
Sulfur, max., %	0.10	0.10	0.10
Corrosion	None	None	None

Motor Fuel – 72 Octane

U. S. Army Specification No. 2-114

E. Detailed Requirements.

E-1 Motor fuel shall be of one type with automatic provisions for locality and climatic conditions, and shall conform to the following requirements:

	Classification A	B	C
ASTM Distillation:			
Maximum Temperature			
for 10% evaporation, F	160	145	130
for 50% evaporation, F	257	245	235
for 90% evaporation, F	356	356	356
Residue, max., %	2	2	2
Vapor Pressure, max. psi (a)	8	10	12
Octane No. ASTM, min.	72	72	72
Gum, mg/100 ml, max. (b)	7	7	7
Sulfur, max., %	0.25	0.20	0.15
Corrosion	None	None	None

The following requirements are common to both specifications:

(a) When vapor-pressure measurements are made on samples taken at the refinery or from tank cars, barges, large tank trucks, or bulk storage, an additional 1 psi in vapor pressure shall be allowed, except that no increase in vapor pressure shall be permitted when the motor fuel is for use in tropical areas (23½ deg. north latitude to 23½ deg. south latitude).

(b) When a finished gasoline contains a non-volatile additive, it shall be permissible to make the gum test on the base stock.

E-2. The class of motor fuel to be supplied in any locality at any one season of the year shall conform to the following tabulation, and each contractor shall automatically make seasonal modifications in accordance with this tabulation:

Territory	Jan.	Feb.	Mar.	April	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.
Section 1	C	C	C or B	B	B or A	A	A	A	A or B	B	B or C	C
Section 2	C	C or B	C or B	B or A	B or A	A	A	A	A or B	B	B or C	B or C
Section 3	B or C	C or B	B or A	B or A	A	A	A	A	A	A or B	B	B or C
Section 4	B	B or A	B or A	A	A	A	A	A	A	A or B	A or B	B

Section 1

Idaho
Iowa
Maine
Michigan
Minnesota
Montana
New Hampshire
North Dakota
South Dakota
Vermont
Wisconsin
Wyoming

Section 2

Colorado
Connecticut
Delaware
District of Columbia
Illinois
Indiana
Kansas
Kentucky
Maryland
Massachusetts
Missouri
Nebraska
Nevada
New Jersey
New York
Ohio
Oregon
Pennsylvania
Rhode Island
Utah
Virginia
Washington
West Virginia

Section 3

Alabama
Arizona[a]
Arkansas
California
Georgia
Mississippi
New Mexico[a]
North Carolina
Oklahoma
South Carolina
Tennessee
Texas[a]

Section 4

Arizona[b]
Florida
Louisiana
New Mexico[b]
Texas[b]

[a]North of 35-deg latitude.
[b]South of 35-deg latitude.

A temperature guide for Grades A, B, and C, where the foregoing zoning does not apply:

A – For mean temperature from 50 F to extreme heat.
B – For mean temperature from 25 F to 70 F.
C – For mean temperature from extreme cold to 45 F

E-3. A tolerance of one week, plus or minus, will be permitted on delivery of the seasonal grades as specified in paragraph E-2.

Appendix 2

OIL, FUEL, DIESEL

for

HIGH-SPEED AUTOMOTIVE-TYPE DIESEL ENGINES

U. S. Army Specification No. 2-102B (Condensed)

E. Detail Requirements.

E-1. This fuel shall conform to the following chemical and physical characteristics:

Class	A	B	X
Cetane No., min.	50	47	45
Distillation Test			
90% point max., F	650	625	600
End point max., F	700	700	650
Flash point min., F	140	115	110
Pour point max., F	0	—20	—40
Cloud point max., F	+10	—10	—30
Kinematic Viscosity, centistokes at 100 F	2.11-4.28	1.8-4.28	1.6-4.28
Carbon residue on 10% bottoms, max.	0.15	0.15	0.15
Sulfur, max., %	1.0	1.0	1.0
Water and Sediment	Nil	Nil	Nil
Corrosion	pass	pass	pass

E-2. The class of fuel to be supplied in any locality at any one season of the year shall conform to the following tabulation and, unless otherwise specified, each contractor shall make deliveries in accordance with this tabulation:

Location	All Year	March through November inclusive	December through February inclusive
Group 1 States (a)	Grade A		
Group 2 States (b)		Grade A	Grade B
Group 3 States (c)		Grade A	Grade X
Territory of Hawaii	Grade A		
Philippine Islands	Grade A		
Canal Zone	Grade A		
Puerto Rico	Grade A		
Alaska	Grade X		
Iceland	Grade X		
Newfoundland	Grade X		

(a) Group 1 states include:

Alabama, Arizona, Arkansas, California, Connecticut, Delaware, District of Columbia, Florida, Georgia, Kentucky, Louisiana, Maryland, Mississippi, Missouri, New Jersey, New Mexico, North Carolina, Ohio, Oklahoma, Oregon, Pennsylvania, Rhode Island, South Carolina, Tennessee, Texas, Virginia, Washington, West Virginia.

(b) Group 2 states include:

Idaho, Illinois, Indiana, Iowa, Kansas, Massachusetts, Michigan, Nebraska, Nevada, New Hampshire, New York, Utah, Vermont.

(c) Group 3 states include:

North Dakota, South Dakota, Minnesota, Montana, Wyoming, Colorado, Maine, Wisconsin.

Appendix 3

OIL, ENGINE, LUBRICATING, ALL-PURPOSE

FOR USE IN AUTOMOTIVE GASOLINE AND DIESEL ENGINES

U. S. Army Specification No. 2-104A (Condensed)

C. Material and Workmanship.

C-1. Engine lubricating oil shall be a refined petroleum product with or without additive agents which shall provide satisfactory lubrication of high-speed, automotive-type gasoline, diesel, or spark-ignition fuel engines when operated under all conditions of service.

D. General Requirements.

D-1. Engine lubricating oil shall be non-corrosive to bearings and engine parts, shall not cause or permit piston-ring sticking or clogging of oil channels, and shall minimize cylinder and ring wear.

D-2. Additive agents, if used, shall remain uniformly distributed throughout the oil at all temperatures above the pour point up to 250 F. If the oil is cooled below its pour point, it shall regain its homogeneity on standing at a temperature of not more than 10 F above the pour point of the oil.

D-3. All engine lubricating oil procured under this specification shall be compatible with all other engine lubricating oils previously procured under this specification.

E. Detail Requirements.

E-1. Physical and Chemical Requirements.

E-1a. Engine lubricating oils shall conform to

Test	Test Limits		
	Grade SAE 10	Grade SAE 30	Grade SAE 50
Viscosity, Saybolt Universal			
Sec. at 130 F	90 to less than 120	185 to less than 255	....
Sec. at 210 F	....	....	93 to less than 104
Viscosity Index, min.	85	55	75
Pour Point, F, max.	0	+10	+15
Flash Point, F, min.	360	390	425

E-2. In addition to meeting the above requirements, oil, engine, lubricating shall be subject to the following additional requirements:

E-2a. Grade SAE 30, Oil, Engine, Lubricating shall pass those engine tests described as Caterpillar Tests Class 1, 2, or 3 in the "Diesel Lubricant Test Manual" as issued by the Caterpillar Tractor Co. and of the latest date approved by the War Department. Grade SAE 30, Oil, Engine, Lubricating shall also pass the test described as "Test Procedure for 500-Hr Lubricating Oil Test," as issued by the General Motors Corp., Detroit Diesel Engine Division, and of the latest date approved by the War Department.

E-2b. Grade SAE 10, Oil, Engine, Lubricating shall pass these engine tests described as Caterpillar Tests Class 1, 2, or 3 specified in the "Diesel Lubricant Test Manual" (Par. E-2a), and shall contain the same additive used in the Grade SAE 30 oils which meet the requirements of the "Test Procedure for 500-Hr Lubricating Oil Test," specified under E-2a.

E-2c. Grade SAE 50 Oil, Engine, Lubricating shall contain the same additive in the same amount as used in the Grade SAE 30 oil as defined under Paragraph E-2a which has been approved for supply by the same manufacturer. It shall also be subject to such engine and/or service tests as the War Department may prescribe.

E-2d. When specified in the inquiry or invitation for bids and in the contract or purchase order, the contractor shall furnish written evidence that the product supplied under the contract has successfully passed the appropriate tests for the grade ordered, prescribed in paragraph E-2a, E-2b, and E-2c.

Appendix 4

PROPOSED FEDERAL SPECIFICATION – No. VV-L-761 (Condensed)

LUBRICANT: GEAR, UNIVERSAL (HYPOID AND OTHER TYPES)

C. Material and Workmanship.

C-1. Universal Gear Lubricant shall consist of a mixture of well-refined mineral oil and a load-carrying additive containing sulfur and chlorine, with or without the admixture of other additives.

C-2. Universal Gear Lubricant shall be satisfactory for the lubrication of all automotive gear units (hypoid or other types), all heavy-duty industrial-type enclosed gear units, steering columns, and fluid lubricated universal joints of automotive equipment, where the sustained operating temperatures of the lubricant are 250 F, or lower.

D. General Requirements.

D-2. Universal Gear Lubricants supplied by different, or the same manufacturers shall be free from any precipitation or turbidity when mixed in any proportions.

D-3. Universal Gear Lubricants supplied by different, or the same manufacturers shall meet the detailed requirements of this specification when mixed in any proportions.

D-4. Universal Gear Lubricants shall contain only those additives in fixed percentages which have been subject to type tests for solubility, effect of rate of shear, and endurance by the National Bureau of Standards or a laboratory approved by them, with the sole exception that materials whose only function is to lower the pour point will not be considered as additives for purposes of this specification.

E. Detail Requirements.

E-1. Universal Gear Lubricants shall conform to the following requirements:

Test	Test Limits		
	SAE 80	SAE 90	SAE 80-140
Viscosity, S. U. at 210 F, sec.	55-65	80-90	115-125
Viscosity Index, Dean & Davis, min.	85	85	118
Channel Test, max., F	—20	0	—20
Load-Carrying Capacity on SAE machine, min., lb.			
Before Heating, 1000 rpm	300	325	325
After Heating, 500 rpm	400	400	400
Shock Test (Chevrolet Bump Test)	Pass	Pass	Pass
Stability Test (100 hr at 300 F)			
Viscosity Increase, max., %	25	25	25
Insoluble Material, max., %	0.5	0.5	0.5
Evaporation loss, max., %	10	10	10
Corrosion Tests			
1 hr at 212 F, Copper Strip	Negative	Negative	Negative
1 hr at 300 F, Copper Strip	Positive	Positive	Positive
24 hr at 200 F, Steel Strip, max., g	0.01	0.01	0.01
Foaming Test, max., ml	150	150	150
Emulsion Test (130 F), Separation of water in 1 hr, min., ml	25	25	25
Turbidity	None	None	None

Appendix 5

TREASURY DEPARTMENT SPECIFICATIONS FOR GREASES

SPECIFICATIONS FOR ITEMS 14-G-1075 to 14-G-1425-400, INCLUSIVE

1. *Technical Requirements.*—The technical requirements for lubricating greases shall be as shown in the table below;

TESTS	TYPE OF GREASE—TECHNICAL TEST LIMITS								
	Type A	Type B	Type C	Type D	Type E	Type F	Type G	Type H	Type I
MINERAL OIL CONSTITUENT:									
Viscosity, S. U. at 100° F., seconds (min.)	100	100	280	280	280	280	280	100	280
Flash point, ° F. (min.)	315	315	350	350	350	350	350	315	350
Fire point, ° F. (min.)	350	350	400	400	400	400	400	350	400
GREASE:									
Work consistency, grade No. (N.L.G.I.)	2	1	1	0	1	0	2	4	3
Penetration, worked (min.)	265	310	310	355	310	355	265	175	220
Penetration, worked (max.)	295	340	340	385	340	385	295	205	250
Soap	(*)	(*)	(*)	(*)	(*)	(*)	(*)	(**)	(*)
Mineral oil content, percent (min.)	80	85	80	90	85	90	75	65	70
Water content, percent (max.)	2.0	2.0	2.0	1.5	2.0	1.5	Trace	2.5	Trace
Ash as sulfates, percent (max.)	6.0	5.0	3.5	3.5	3.5	3.5	5.0	8.0	7.5
Free alkalinity (calculated as NaOH), percent (max.)	.2	.2	.2	.2	.2	.2	.2	.2	.2
Insoluble matter, percent (max.)	.10	.10	.10	.10	.10	.10	.10	.10	.10
Dropping point, ° F. (min.)							250	250	300
Corrosion	Neg.	Neg.	Neg.	Neg.	Neg.	Neg.	Neg.	Neg.	Neg.

* Soda, lime, or aluminum, or mixture of same.

**Lime, aluminum, zinc, or lead, or mixture of same.

2. *General Requirements.*—Each type of grease shall be a smooth homogeneous mixture of refined mineral oil and pure soap of the kind specified, completely saponified; shall contain no fillers such as resin, resin oils, talc, wax, powdered mica, sulfur, clay, asbestos, rubber, or other undesirable or deleterious impurities; and shall have no objectionable odor.

3. *Special Requirements.*—Greases, types C, D, E, F, G, H, and I shall be water-repellent in service; types C and D shall be suitable for use in pressure guns; types E and F shall be "adhesive", "stringy", and "tacky" in character, and shall be suitable for lubrication of the tractor rollers on continuous-track-type tractors.

Appendix 6

CONDENSED SUMMARY OF ARMY SPECIFICATIONS FOR GREASES

May 1, 1942.

Detail Characteristics					Heavy-Duty
Grade No.	0	1	2	Water Pump	Wheel Bearing
Army Specification No.	2-106	2-107	2-108	2-109	2-110
Worked Penetration at 77 F	355 to 385	310 to 340	265 to 295	175 to 205	220 to 250
Oil Content, min., %	90.0	85.0	82.0	65.0	70.0
Water Content, max., %	1.5	2.0	1.5	2.5	Trace
Ash as Sulphates, max., %	3.5	3.5	6.0	8.0	7.5
Free Alkali (as NaOH), max., %	0.2	0.2	0.2	0.2	0.2
Insoluble Matter, max., %	0.10	0.10	0.10	0.10	0.10
*Free Fatty Acids (as oleic), max., %	*0.30	*0.30	0.30	..	0.3
Dropping Point, min., F	..	..	250	210	300
Corrosion	Nil	Nil	Nil	Nil	Nil
Mineral Oil					
Viscosity at 100 F, min., sec.	280	750	..	100	..
Viscosity at 210 F, min., sec.	..	..	75	..	75
Pour Point, max., F	..	..	+5	..	+5
Flash Point, min., F	350	350	350	315	350
Soap, Type of	Note 1	Note 1	Note 2	Note 3	Note 2
Fillers					

*Omit determination if aluminum soap is present.

Note 1. Base shall be a sodium and/or calcium and/or aluminum soap of one or more of the higher fatty acids.

Note 2. Base shall be a sodium and/or calcium and/or barium soap of one or more of the higher fatty acids.

Note 3. Base shall be a calcium soap of one or more of the higher fatty acids.

May 13, 1942.
Engineering Division,
Fuels & Lubricants Branch.

Appendix 7

LUBRICATION GUIDES

a. Lubrication guide for passenger cars, motorcycles, trucks, tractors, and chassis furnished by the Quartermaster Corps.—Lubricants used as listed below must conform to specifications indicated.

Vehicle make	Engine: Oil, engine, all purpose, Grade: Winter, 0° F. to 10° F.	Winter, 10° F. to 32° F.	Summer, 32° F. to 90° F.	Summer, Over 90° F.	Transmission and transfer case, front and rear axle differentials, steering gear and winch gears: Lubricant, gear, universal grade No., Below 0° F.	Above 0° F.[1]	Propeller shaft splines and universal joints, chassis fittings, universal joints at front wheel hubs: Grease, general purpose grade No., Below −10° F.	−10° to 32° F.	Above 32° F.	Wheel bearings: Grease, general purpose, All temperatures[2]	Shock absorbers
American Bantam	10	10	30	30	80	90	0	0 or 1	1	2	None.
Autocar	10	10	30	50	80	90	0	0 or 1	1	2	Light.
Chevrolet	10	10	30	30	80	90	0	0 or 1	1	2	Do.
Corbitt	10	30	30	50	80	90	0	0 or 1	1	2	None.
Diamond T	10	30	30	50	80	90	0	0 or 1	1	2	Heavy.
Dodge, Plymouth	10	10	30	30	80	90	0	0 or 1	1	2	Light.
Federal	10	10	30	50	80	90	0	0 or 1	1	2	Do.
Ford	10	10	30	30	80	90	0	0 or 1	1	2	Heavy.
G. M. C	10	10	30	30	80	90	0	0 or 1	1	2	Light.
International	10	30	30	50	80	90	0	0 or 1	1	2	Do.
Mack	10	10	30	50	80	90	0	0 or 1	1	2	Heavy.
Packard	10	10	30	30	80	90	0	0 or 1	1	2	Light.
Studebaker	10	30	30	30	80	90	0	0 or 1	1	2	Do.
White	10	30	30	50	80	90	0	0 or 1	1	2	Do.
Willys	10	10	30	30	80	90	0	0 or 1	1	2	Do.
Harley-Davidson	10	30	50	50	Eng.		0	0 or 1	1	2	None.
Indian	10	30	50	50	Oil[3]		0	0 or 1	1	2	Do.

[1] Under extreme temperature conditions (over 90°F.) and where leakage is experienced due to defective seals, lubricant, gear, universal grade 80-140, may be used in place of grade 90 as a temporary expedient.

[2] Under extreme temperature conditions (over 90° F.) when wheel bearing grease leakage is experienced due to defective seals, grease, wheel bearing, heavy duty, may be used as a temporary expedient.

[3] For temperatures below zero, engine oil may be diluted with 10% or more of kerosene or diesel fuel.

b. Lubrication of miscellaneous items at all temperatures.

Generator and starting motor	Same as engine
Fan and distributor	Do.
Water pump	Grease, water pump
Oil bath air cleaner	Same as engine
Fifth wheel	Same as chassis
Brake fluid	ISO, Delco 9, Puritan, or Lockheed 21
Shock absorber fluid	Light-Delco Heavy-Houdaille
Motorcycle chains (all services)	Engine oil
Vacuum booster cylinders	SAE 10 (all seasons)
Hydraulic dump hoists	Do.
Speedometer drive cables	Grease, general-purpose, No. 0 or No. 1

c. Lubricant designation, specification, and purchasing guide.

Type	Grade No.	Specification designation
Grease, general purpose	0	U. S. Army No. 2-106.
Do	1	U. S. Army No. 2-107.
Do	2	U. S. Army No. 2-108.
Grease, water pump	4	U. S. Army No. 2-109.
Grease, wheel bearing, heavy duty.	3	U. S. Army No. 2-110.
Lubricant, gear, universal	All	Federal Specification VV-L-761.
Oil, engine, lubricating, all-purpose.	do	U. S. Army Specification No. 2-104-A.

CONVERSION for WAR–
Influence of Automotive

by JOSEPH GESCHELIN

Detroit Technical Editor, Chilton Co., and SAE Vice President representing Production Engineering

CONVERSION of the automotive industry, in one sense, began about the middle of 1940 and accelerated during 1941. Pearl Harbor found the industry well on its way to all-out conversion to war production. Some idea of the degree of participation in the war effort may be gained from the statement issued by General Motors Corp., in a full-page advertisement in the newspapers on April 13, 1942. Under the heading – "Here's What We Have Accomplished" – GM lists the following contracts in hand:

One-third of the machine guns.
More than one-third of the Army trucks.
More than half of the Navy's diesel engines.
Two-fifths of the aviation engines.
More than one-fourth of the tanks.
Three separate types of complete airplanes.
Numerous parts for planes produced by others.
Guns and gun parts of many sizes, from 0.30 caliber to 5 in.
Shell and cartridge cases.
Over 1000 orders and contracts for war materials.

If space permitted, it would be quite to the point to outline the war production programs of other manufacturers. Thus Chrysler has become the major producer of medium tanks for the Army, is making Army trucks, Bofors guns, marine engines, ordnance of various kinds. Ford is making motor trucks, jeeps, airplane engines, and is rapidly ap-

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IN a discussion of the difficulties that are being overcome in converting automotive plants to the production of war equipment with virtually no new machine tools, Mr. Geschelin emphasizes the difference between the production requirements of an automobile engine and a modern airplane engine by pointing out that:

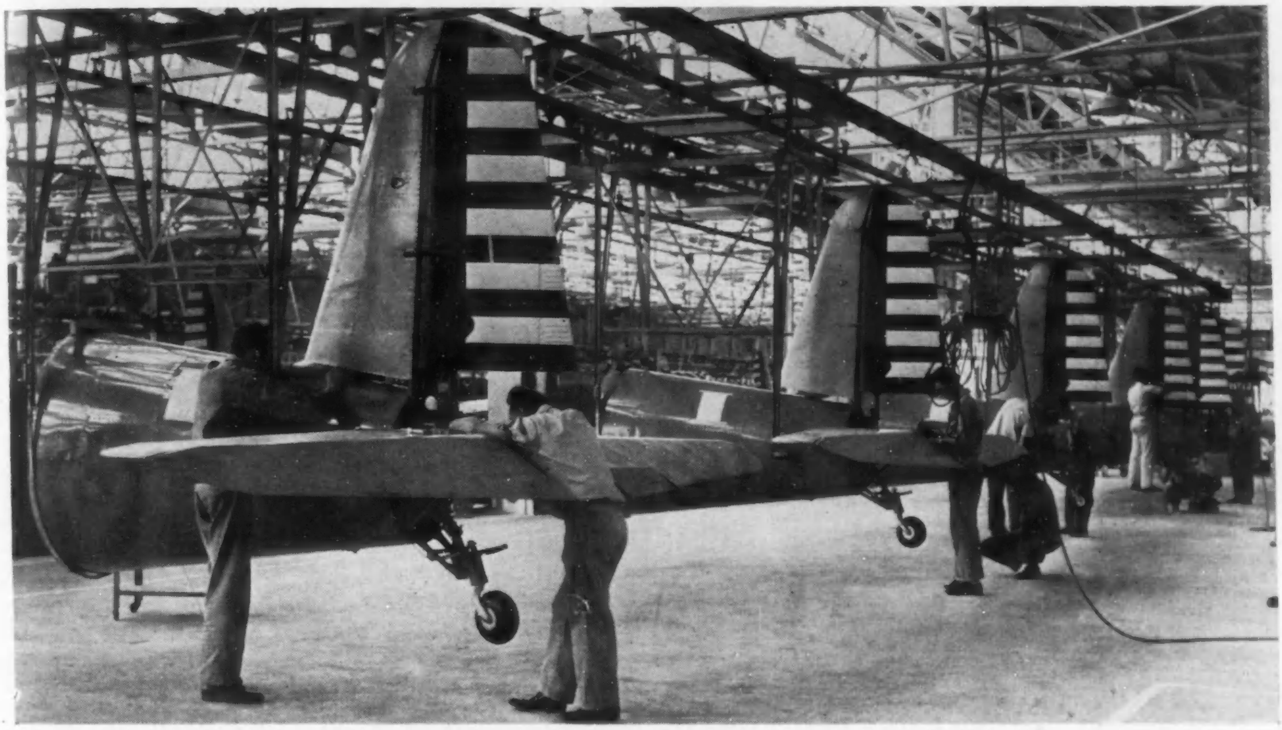
The connecting rod of the Allison airplane engine requires 93 operations compared with 25 for the Cadillac passenger-car engine; the Allison crankshaft requires 80 operations, the Cadillac crankshaft 62; and the Allison cylinder block has 17 pieces compared with a single block for the Cadillac engine.

After calling production know-how, "the major contribution of the automotive industry," Mr. Geschelin draws attention to the fact that the automotive industry also is largely responsible for bringing American production machinery to its present high state of development. Without the automotive industry's demand for mass production and encouragement of invention, he contends, the machine-tool industry would never have reached its present stage of development and expansion. He goes on to show definitely how the industry has fostered advancement in machine tools, metallurgy, materials handling, surface finish, gear practice, heat-treatment, and shot blasting.

In the remainder of his paper the author explains – naming product, company, process, and machine – how automotive mass-production methods are influencing the production of such war equipment as airplane engines, military airplanes, Ordnance materiel, and military vehicles.

THE AUTHOR: JOSEPH GESCHELIN (M '21) became Detroit technical editor of all Chilton publications in 1936. Upon graduating from Cooper Union Institute of Technology with a B.S. in Mechanical Engineering, he gained experience in jobs in motor truck and parts plants that combined practical engineering and sales promotion, which ultimately prepared the way for technical editorship. Present SAE vice president representing the Production Activity, Mr. Geschelin is also consultant to the Bureau of Ships, U. S. Navy.

Mass-Production Methods



Tail assembly section – one of the highly mechanized departments in the Vultee plant – is typical of the conveyorized operations here

proaching full production on the huge Consolidated bombers at Willow Run. Packard is making engines for the Navy, Rolls-Royce airplane engines, and other items. Studebaker is producing airplane engines, motor trucks, etc. Willys is making jeeps and shells. Space does not permit a full inventory of the tremendous expansion of war production undertaken by the parts makers and passenger car builders.

Initial stages of conversion were represented by expansion of facilities, by the erection of new plants, and by the acquisition of entirely new equipment. More recently, the process has been extended to the utilization of the existing factory buildings, the using of available equipment supplemented with new machinery.

We have reached the stage now where there is no longer the time for new facilities of any kind. From now on, the industry must convert all of its existing buildings and must find ways to use all available machinery by the tricks of improvision of which its production men are past masters. This process has been taking shape for months, and before long many of the industry's plants will have all but lost their identity as producers of motor cars and certain kinds of parts.

With the shortage of capacity for the manufacture of all of the machine tools that are needed for the economical production of the weapons of war, the ingenuity of master mechanics and production men must be taxed to the very limit in finding ways to utilize existing machinery even though it is recognized that that is not the best way to do the job. Time and not cost is of the essence in war.

But the implications of "conversion" have been grossly misunderstood by laymen, by the man on the street, and even by people whose proximity to the production process should give them a better appreciation of what is involved in the conversion from the manufacture of automotive products to the manufacture of the weapons of war.

There has been a mistaken notion that an automobile plant, for example, could be utilized without much change to the immediate production of airplane engines, of bombers, or of fighters. The layman has failed to understand that the facilities of an individual plant must be surveyed to determine exactly what kind of war products can best be produced; that the industry must gear up to produce

[This paper was scheduled for presentation at the 1942 Semi-Annual Meeting of the Society, which was ruled out because of transportation priorities.]

an endless variety of such products; and that actually only a small percentage of available equipment can be turned to useful purpose except on specific tasks for which it is best suited.

For instance, one of the GM units is making Oerlikon guns. This operation requires about 85% of the equipment in the form of heavy-duty milling machines. No single plant of the corporation has that many milling machines in its normal function of building motor cars or parts. About 12% of the equipment, consisting of drill presses and grinders, was salvaged from a motor-car plant for this purpose.

Another example of confused thinking – and this has had a serious effect on the thinking of the country – is the mistaken notion that a passenger-car engine line can be converted to the production of a modern airplane engine. While it is true that the major components of both engines bear similar names and, in general, are machined on equipment bearing similar names, neither the parts nor the machines have anything else in common. Airplane engine parts not only are larger and made of materials more difficult to cut, but the requirements of accuracy and extremely fine surface finish demand larger and more massive machine tools. Moreover, more machines are required for a relatively smaller volume of production, as may be noted from the following examples in connection with the Allison engine: Compared with the Cadillac passenger-car engine – the connecting rod requires 93 operations as compared with but 25; the airplane-engine crankshaft requires 80 operations as compared with 62; the crankcase has 17 individual pieces as compared with a single block for the Cadillac engine.

■ Production "Know-How"

Essentially, the major contribution of the automotive industry is its "know-how." This has made it possible to place on a production basis the manufacture of the many vital things that heretofore were made by hand or in relatively small quantities.

For over a decade the automotive industry has been accelerating the tempo of mass-production techniques and originating unique processes which have made it possible to build better and faster, at constantly lower cost levels. We all have realized that it was a task well done. But it has taken a grave war emergency to show the country how far-reaching these accomplishments have been.

In appraising these advances, it is not sufficient to point to the achievements of mass-production nor to the development of interchangeability on a vast scale. The real contribution lies much deeper. For example, in the process of perfecting advanced methods, the industry has encouraged the invention and manufacture of marvelous production machinery. This demand, in turn, has permitted a wholesale expansion of the basic machine-tool industry.

Too, there has been a parallel development of improved cutting tools, of improved foundry practice and forging practice, of radical procedures in the field of metallurgy. In fact, all of the manifold activity that goes into the automotive production process has been fostered by our industry – the greatest single consumer of such services.

The ramifications of modern production methods are so broad and take in so much territory that it would be quite difficult to touch upon individual examples in word pictures. Accordingly, I have chosen to present a rather generalized perspective of the situation and which can best be visualized by reference to a large group of illustrations touching upon a variety of problems.

Let us consider an outline of technical progress which constitutes the backbone of the wherewithal for making the weapons of war:

■ Technical Progress Reviewed

1. *Machine tools* – As a direct result of the needs of mass production, there was available the host of equipment so essential in defense production – precision boring machines, precision thread grinders, multiple-spindle drilling, tapping, and boring machines, precision grinders of every type, surface broaching machines, automatic lathes and turret lathes of various kinds, heavy-duty precision milling machines, hydraulic presses, precision honing equipment, and so on.

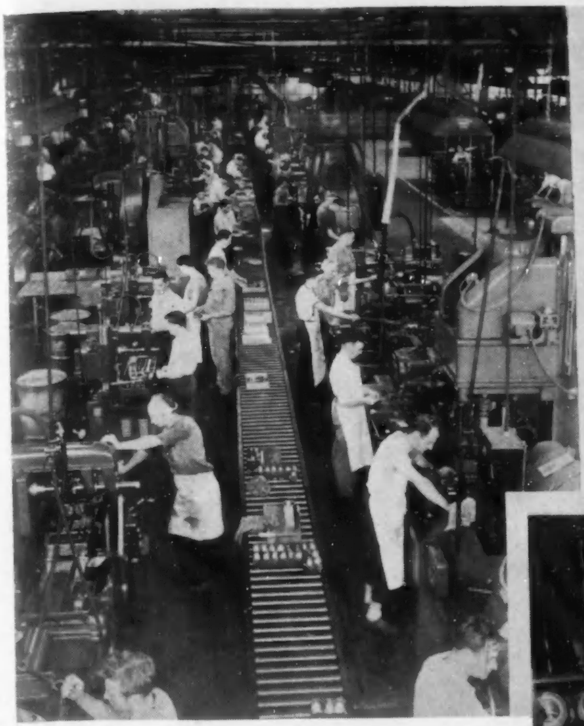
2. *Metallurgy* – Because the automotive industry demanded and encouraged the development of the finest ferrous and non-ferrous metals possessing exceptional physical properties, durability, and good machinability, it has been possible to make the weapons of war more powerful, more durable, and with mobility hitherto unknown.

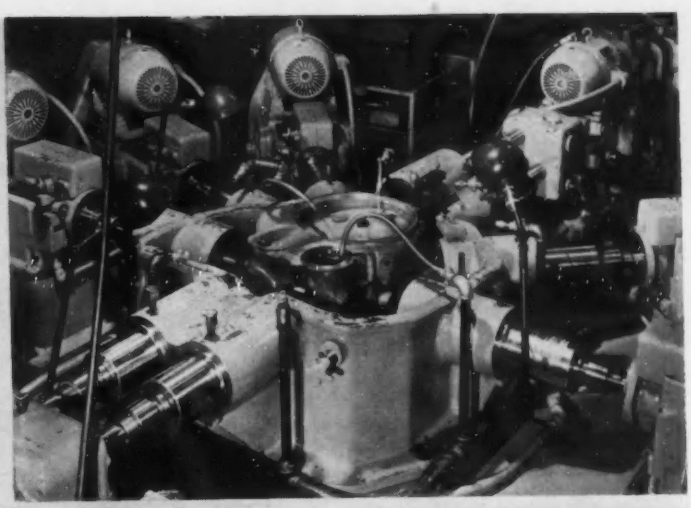
3. *Materials Handling* – Mass-production principles have developed the science of materials handling, have made available the conveyor systems, the power cranes and hoists, the industrial trucks, that are being exploited to the fullest extent in facilitating the smooth flow of war products on the machine lines.

4. *Surface Finish* – Engineering talents and production techniques have been bent for a long time to the requirements of fine surface finishes for certain parts of the mechanism. This has been related to the wider uses of precision boring machines, precision grinders, honing machines, lapping machines, superfinish equipment, and so on. And it has been supplemented with instrumentation such as the Abbott Profilometer, the Brush Surface Analyzer, Magnaflux, P & W Electrolimit gage, Sheffield gages, and so on. These techniques and instrumentation have proved to be of inestimable value in the defense effort, particularly in the production of aircraft parts where surface finish is so important. Too, these things have made it feasible to widen the horizon of subcontracting since instrumentation makes it possible to work according to reproducible standards.

5. *Gear Practice* – The requirements of "silent" gear trains, of durable gears of uncommonly small physical size, of long-lived highly stressed gears, that have dominated the thinking of automotive experts during the past ten years have resulted in the availability of methods and equipment without which it would be almost impossible to produce the aircraft gearing in large volume. Improved methods of gear shaping and hobbing, shaving and lapping, precision grinding where needed, and allied techniques all contribute to the steady increase in the production of the finest aircraft gearing known to the art.

6. *Heat-Treatment* – High-frequency induction hardening as exemplified by the Budd process, by Tocco, and by





(Above) View of the bearing-cap line – one of many highly mechanized departments in the Packard Rolls-Royce operation

(Top right) Special W. F. & John Barnes horizontal multiple-head drilling machine on Allison-engine blower section

(Bottom right) View in Continental plant – cylinder-head department – with Ex-Cell-O precision boring machine in foreground; Kearney & Trecker vertical milling machine in rear

certain furnace manufacturers has greatly speeded the selective hardening of wearing surfaces. We also may point to the utilization of infra-red baking lamps, now so widely employed for the rapid drying of paint. In one instance – at Aircooled Motors Corp. – infra-red lamps are being used to heat parts for interference fit assembly. Another modern technique is that of flame-hardening for selective surface-hardening effects. This process, too, has been placed on a mass-production basis.

7. *Shot Blasting* to prevent fatigue resistance – This is an advanced method that deserves the attention of all engineers and production men. It has been employed for some years as a sort of salvage operation in the case of axle shafts. Wherever the axle shaft is overloaded and an additional factor of safety is required, shot blasting has provided the means of extending shaft life. More recently, the method has been employed in the finishing of the special dished clutch springs used by Chevrolet and Buick. However, the recent shift to ferrous pistons at Buick has provided the setting for a wider use of this principle. Due to the added inertia loading, it was felt that the connecting rod would have to be strengthened. Instead of resorting to a heavier rod, with its attendant heavy expenditure for new die equipment, Buick switched to an alloy steel, supplemented with shot blasting. This has solved the problem quite effectively. And in this is a lesson for us all. In the future, shot blasting should be properly considered as a special production process in the same sense as heat-treatment, for example. Through its use, it may be feasible to make parts smaller and lighter than indicated by mathematical analysis.

From this generalized picture, let us turn to some actual examples of mass-production methods in various areas in which weapons of war are being made.

■ Airplane Engines

Beyond a doubt the greatest influence of automotive production methods has been felt in placing aircraft-engine building on a mass-production basis. True enough, the leading engine builders in this field had had many years of

experience and had established standards for materials, for surface finish, for tolerance, and so on. But the translation of these standards on a volume basis was entirely a new development.

■ In-Line Engines Bring Problems

Another type of production problem was presented with the entry of in-line liquid-cooled engines. Although in detail the in-line engine is subject to the same standards of manufacturing, it has numerous peculiarities quite foreign to the building of radials. As examples of this difference, we may note the machining of long slender parts such as the cylinder head, crankcase sections, camshafts, crankshafts, and propeller shafts. Such parts demand special care in handling if the dimensional accuracy is to be preserved and if distortion is to be eliminated.

For one thing, it means that the number of cuts and machine set-ups must be increased. This in turn requires more machinery, more fixtures, more rigid fixtures and tooling, more massive machines.

Generally speaking, one common characteristic of all aircraft-engine building is that of extremely heavy chip removal. It is common practice to find that the ratio of rough casting or rough forging weight to the finished weight is of the order of five to one.

Some idea of the magnitude of the production problem may be gained from the following basic statistics on the Wright Cyclone 14:

1. The engine is composed of 3500 different parts, totaling 8500 individual pieces.

2. The estimated number of machining operations is placed at 80,000.

3. The quality control program anticipates some 50,000 inspection operations.

To handle this volume of machine-shop activity just on the parts to be made in this plant entails the installation of 2400 machine tools of every description, including the most advanced equipment known to the art.

Easily one of the most spectacular pieces of equipment in one of the Wright machine shops is the fully automatic process machine developed by Greenlee for the machining of cylinder heads. With 14 heads per engine there is an opportunity to match the automaticity and productivity of the best examples to be found in passenger-car practice. This is a special transfer machine for drilling, countersinking, reaming, and tapping. It is made up of two sections, extending about 80 ft in length in a line-up of coordinated unit-type heads. It is estimated that this machine replaces about 40 machine tools of conventional type.

The first unit of this Greenlee machine has 16 stations, taking some 25 different operations. The second unit has 57 stations, performs 46 operations.

■ Military Airplanes

When it comes to the making of airplanes, that's the sphere of the specialists who have been at it since the last war. Nevertheless, it is obvious that the mass-production techniques evolved in the motor-car industry have exerted a profound effect upon the manufacture of military airplanes. Doubtless much of this transfer of information came about through the medium of seasoned production men who cast their lot with the airplane industry at the start of the early defense program.

Make no mistake – the airplane industry is big mass-production business today. And I can tell you this from first-hand experience gained by actually visiting the plants on the West Coast. Here will be found the very latest types of familiar makes of metal-cutting equipment in tool rooms and in machine shops, huge presses larger in physical size than most presses found in auto-body plants, albeit of lower pressures; latest types of water-back spray booths larger than anything we have seen elsewhere – large enough to accommodate the largest bomber without cramping.

■ Assembly Lines Conveyorized

When it comes to materials handling there are industrial trucks, monorail conveyors, cranes and hoists, floor conveyors – all of the tricks known to the art. But particularly impressive are the mechanized assembly conveyors. Airplanes are being assembled on moving conveyor lines with overhead monorail drops for engines, for complete fuselage sections, for wing sections, and so on. Perhaps the most imposing of these is the assembly building for the Consolidated B-24 bombers, among the largest of the four-engined ships made in this country. The assembly building houses a U-shaped conveyor line 1500 ft in length. The bomber starts with the fuselage on one arm of the line, continues along 56 stations, winds up at the end of the other arm ready to take to the skies, after final touches, tune-up, and inspection.

Too, the airplane industry has developed new techniques which may prove to be of value to other industries: At Lockheed, a unique method for the rapid X-ray examination of castings on a really mass-production basis – 20,000 pieces per day. At Douglas the introduction of the Guerin process of producing stampings with rubber dies. In others, the development of fast and accurate methods of reproducing templates for stamped and formed parts by photo processes and by a new X-ray process.

Peering behind the scenes it is obvious that the overnight transition to mass production may be credited to the spirit of cooperation and unity between otherwise competing manufacturers in a time of national emergency. The fact is that many tricks and subtleties of fabrication of aircraft structures have been known and practiced in individual plants. These were their stock in trade, representing a margin of superiority when business was as usual. Today these techniques have been freely exchanged in the interest of the common good.

The West Coast plants are proficient at welding, having had plenty of experience with the welding of aluminum and tubular structures. This was the proving ground for the remarkable Sciaky resistance welders. And today the largest batteries of these controlled-cycle welders are found on the Coast. More recently these have been supplemented with specialized welders built by Federal, Acme, Taylor-Winfield, and others. Every trick of the welder's trade is employed – the oxy-acetylene torch, electric arc, spot welders, butt welders, seam welders.

Before the airplane producers were able to take advantage of mass-production press shop methods, every plant relied upon the use of the drop hammer. During the past year, drop-hammer dies have been made of Kirksite, a special high-zinc-content alloy. We had heard of this

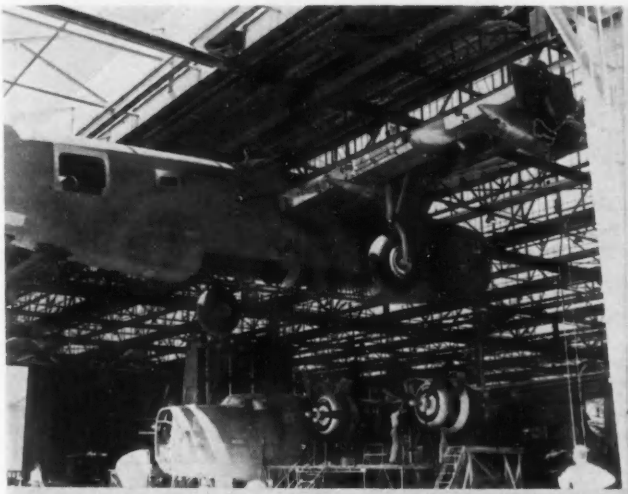
activity but did not realize that this meant that each plant owned and operated its own zinc foundry with from one to three large melting furnaces, served by pattern shops, model shops, and provisions for making plaster forms for casting the dies. And with a yearly consumption of thousands of tons of zinc metal, most of it reclaimed over and over again.

However, with the mounting tide of airplane production, we find the entry of huge drawing presses bearing names familiar in the automotive field – Clearing, Hamilton, Birdsboro, Bliss, Watson-Stillman, Hydraulic Press Mfg. Co., and others. The older drop hammers are giving way to the modern hydraulic press, although all of the available equipment will continue to be employed in the interest of getting the job done in a hurry. Several press shops boast large batteries of Chambersburg Cecostamp presses which constitute the most modern version of a sensitive but extremely powerful successor to the older drop hammer.

Another important newcomer in the field of metal stretching is the Erco "stretching press." Douglas and several others have installed this equipment recently. Instead of forming large skin panels in big presses with bulky dies, the aluminum sheet is stretched over a wood form. The ends of sheet are clamped in pneumatic holddowns on fixed tables flanking the center station of the machine, while bent loosely over the form. Then the center section is slowly inched upward to the end of a predetermined stroke, deforming the sheet into the desired contour. Fast and simple, it represents a major saving in time and expense.

The very latest development along this line is a 150-ton metal stretching press placed on the market only a few weeks ago by the Hydraulic Press Mfg. Co. This is an unusually large machine, capable of stretching large panels of steel or aluminum for airplane sections and, in general, for many short-run large sheet metal parts. The use of the H-P-M stretching press will expedite such operations by eliminating the use of large costly dies.

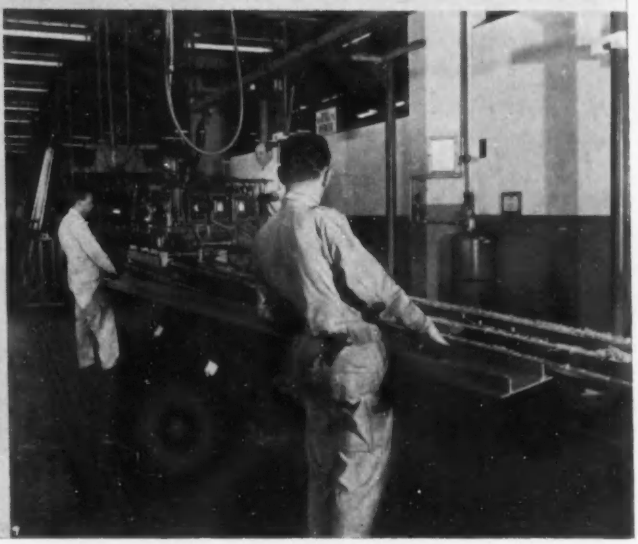
With the passing of time the airplane producers and their subcontractors will be able to introduce still further refinements in production methods, in special techniques for conserving precious time. One recent development along this line is the widening use of the du Pont explosive rivet for blind fastenings. This has proved popular and



(Top) Big four-engined Consolidated bombers are toted out to final assembly stations on the overhead crane system

(Below) Typical of big welding departments on West Coast is this one in the Vultee plant, featuring Sciaky welders

(Top) Douglas bomber wing center sections are built up on this conveyorized assembly line



(Below) Onsrud special milling machine developed for Lockheed, milling long extrusions for interceptors, saving more than three days' time per ship

valuable as a means of reducing the time for joining inaccessible sections, not only speeding up the assembly but materially reducing the cost and assuring a safer blind fastening.

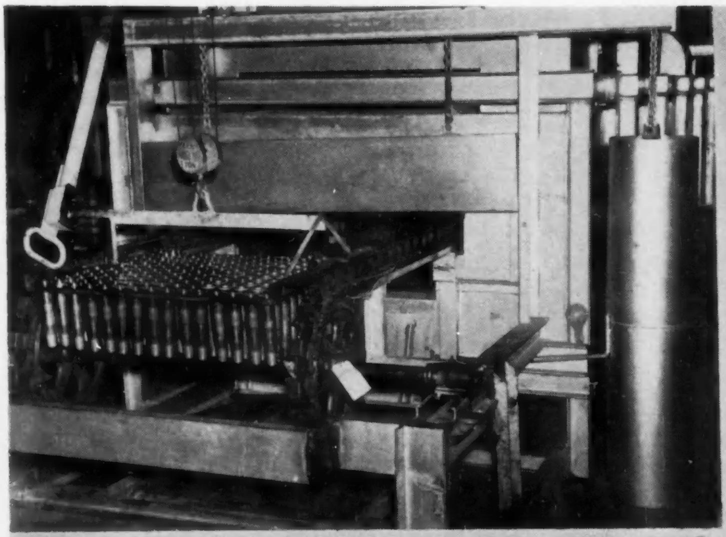
Among the new production techniques at Lockheed is the Onsrud high-cycle extrusion milling machine, developed for machining the ten long extrusions required for a single P-38 ship. Net result of the introduction of this method is a saving of $3\frac{1}{2}$ days of production time per ship. Prize exhibit in the press shop is a new Watson-Stillman hydraulic press of 4500 tons capacity.

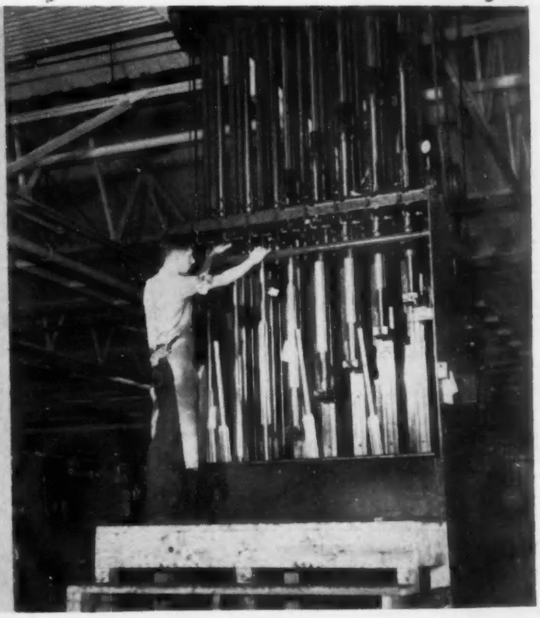
■ Ordnance Materiel

When it comes to items like machine guns, anti-aircraft guns, shells, cartridge cases, and so on, which are being made in quantities hitherto unknown, the pressure of volume and the production-wise experience of automotive factory executives have combined to revolutionize the art. It must be remembered that during several decades of peacetime activity, the making of ordnance materiel has been the responsibility of the Army arsenals. Ordnance specialists have been developing and improving the design of weapons and ammunition, have been creating standards and specifications, but they had little opportunity to experiment with new methods of manufacture, particularly in the field of mass production.

Consider the experience of General Motors with its machine-gun contract. At the start, experimental production and training of workers was done with old equipment salvaged out of the arsenal stores. But, as the picture developed and as the skilled production men began to study the problem, they found many ways in which machine guns could be built quicker and better and at lower cost than ever before. Fortunately, in these initial studies, GM was encouraged and assisted by Ordnance officers, permitted to make departures from established practice of long standing.

Out of this came some striking contributions adapted directly from automotive experience. One example was the





(Above) Eight-barrel W. F. & John Barnes vertical reaming machine is used in General Motors plant for rifle-reaming Oerlikon gun barrels – Earliest development along this line was a 6-cyl drilling machine for gun barrels in General Motors

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(Top left) Electric Furnace Co. drawing furnace in use at Guide Lamp Division, for stress-relieving of small cartridge cases

●

(Bottom left) National and Ajax upsetting machines in use at Olds for forging shell blanks. An Ajax induction heating furnace may be seen at the right

introduction of the W. F. & John Barnes vertical, six-spindle rifle drilling machine which handles six barrels at a time and increases the output of barrels many-fold. This was followed by a study of the application of Carboloy-tipped tools for rifle-drilling, promising not only a further increase in productivity but longer tool life and less interruption of machine output due to tool grinding.

Similarly, there came the introduction of the twelve-spindle, vertical Baush rifle-reaming machines, designed to expedite the reaming of barrels. Still another major advance was the development of a horizontal broaching machine by Illinois Tool Works, for producing barrel rifling by broaching. This procedure cuts the rifling time to but a fraction of previous practice.

■ Broaching Widely Employed

It is our understanding that rifling by broaching stems from work done by the government arsenals some years ago, paving the way to mass-production set-ups under present conditions. One of the most recent applications of this technique is the big American horizontal broaching machine used for rifle-broaching the Oerlikon gun barrel. Here the job is done in a space of but a few minutes using a series of four individual broaching tools.

Surface broaching has been adopted for a number of detail operations, greatly simplifying such operations. Multiple-spindle drilling and tapping machines, heavy-duty milling machines, and other types of equipment familiar in automotive plants have been applied in an effort to speed output.

Materials handling is quite conventional, taken right out of the book of rules of the GM plants.

Another innovation was the use of Baird tumbling barrels lined with Neoprene for tumbling the small components of the machine gun to remove burrs, and to break sharp edges. This method eliminated the conventional practice of hand filing and grinding and polishing – again saving time and labor.

Budd and GM furnish good examples of advanced practice in making shells in huge quantities. Each one features modern production equipment familiar in automotive plants. Both use induction heating to speed up the heating of the billets – Budd using its unique induction process, GM using Ajax induction furnaces. At GM will be found compact machine lines reminiscent of passenger-car practice, with monorail conveyors for transporting billets to the forge departments, for transporting finished forgings to the machine shops, and through the machine lines.

At GM the big shell forgings are cold-nosed on a huge press. This is probably the first time that the cold-nosing procedure on large shells had a trial in production and it has worked out admirably.

The making of cartridge cases is an old art. In fact, the steps in the procedure remain unchanged even in mass production. But the equipment and the layout as it was developed at a GM plant, for example, were brand new and embody a major forward step in productivity and quality. Here will be found the familiar Bliss and Toledo presses with multiple indexing dies, automatic lathes, Lindberg furnaces, and so on. For large cartridge cases, they have installed huge hydraulic presses built by Hydraulic Press Mfg. Co. Materials handling has been worked out along the lines of automotive practice, making the department extremely compact and most efficient in the use of time and labor.

Another innovation is the removal of scale by sandblasting, using the Pangborn equipment so widely employed throughout the industry for cleaning forgings and castings.

No matter what item of ordnance materiel has been undertaken by an automotive producer, its processing and treatment have been worked out along the lines of automotive parts production, utilizing equipment which has given a good account of itself in our industry.

■ Military Vehicles

When it comes to the building of military vehicles and bogies, the industry can point with pride to the performance of one of the prominent axle manufacturers, whose engineers have worked with the Army for the past 20 years in preparation for the emergency. In consequence, axle design has been progressively improved and harmonized with the needs of military vehicles. Today, this manufacturer has swung most of its production facilities to the making of axles and bogies for military use, and is building these products with the same methods and substantially the same equipment as has been employed for civilian products of the same character. Perhaps the chief change lies in the fact that this manufacturer has found it necessary to bring in additional equipment so as to increase capacity.

In much the same manner, a prominent axle and transmission manufacturer has turned all of its facilities to the needs of the war effort, making axles, light tank transmissions, half-track vehicle transmissions and transfer cases, universal joints, and so on. In the main, this manufacturer is employing its original equipment for the manufacture of war products, supplemented with new tooling wherever required. However, in the case of the huge tank transmission, it was necessary to acquire many new items of equipment such as the big radial drills, planers, and lathes.

Eaton and other suppliers in this field also have taken on orders for products quite similar to their regular line of parts, thus making it possible for them to shift onto war work without delay and without waiting for new machinery.

■ Summary

In conclusion, it may be said that automotive manufacturing practice has had the effect of dominating the picture of war production. Even a cursory examination of the methods and equipment employed in the production of items of the kind we have touched upon, reveals machinery made familiar in motor-car plants throughout the industry. It is safe to say that the availability of such equipment and the development of specialized techniques have been responsible, in the main, for the rapid expansion of war production and for the relative ease with which the weapons of war have been tooled for mass production.

By the same token, many of the new developments in machine-shop management resulting from the war effort will constitute the base for the advanced practice of the future. In this respect, the current activity will have its effect upon improvements in automotive practice after the emergency is out of the way.

HEAVY-DUTY LUBRICATING OILS

HEAVY-DUTY lubricating oils containing metallo-organic compounds were effective in decreasing, but not totally preventing, piston-ring pinching and sticking in Naval diesel engines, Mr. Klemgard reports, summarizing the results of tests on medium and high-speed two and four-cycle diesel engines of both automotive and heavy submarine types, in which various additive and non-additive type oils were tested.

In general, he continues, one or another of the additive-type lubricating oils was superior to non-additive type oil in piston lacquering, piston-ring sticking, and wear of liners and piston rings in the test engines. The desired objective, he concludes, is a combination of the best diesel engineering designs and heavy-duty lubricating oils of the most universally favorable characteristics.

Author's Note: The opinions or assertions contained herein are the private ones of the author and are not to be construed as official or reflecting the views of the Navy Department or the Naval Service at large.

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THE AUTHOR: EDWIN N. KLEMGARD (M '30) is the inventor of a cold treating process for removing sulfur from cracked naphthas without appreciable polymerization and knock rating losses, used by Standard Oil Co. of Calif. He has also taken out many patents on lubricating oil and grease compositions, and is author of books and technical articles on lubrication. Now senior petroleum technologist, U. S. Naval Engineering Experiment Station, Annapolis, he draws upon long commercial-company experience in the lubricating field. Mr. Klemgard received a B.Sc. degree from Washington State College in 1922.

THE Bureau of Ships ad Interim Specification 14-0-13 (INT), covering the requirements of lubricating oils for Naval diesel engines, was issued May 1, 1941. This interim specification includes chemical laboratory tests, laboratory diesel-engine tests and, when practicable, trials in diesel engines in the Naval service by the Forces Afloat. Before a test is authorized, evidence must be submitted showing that the oil has been successfully used in commercial diesel engines for at least six months. In the tests leading to an approval of a suitable heavy-duty oil, the results obtained in diesel engines are stressed. The progress made in the establishment of test procedures is shown in this paper.

Grades – Diesel-engine lubricating oils are classified as Navy Symbol 9000 series oils. The last three digits represent the approximate Saybolt Universal viscosity at 150 F, that is, N.S. 9250 oil is a diesel-engine lubricating oil of approximately 250 sec Saybolt Universal Viscosity at 130 F. Grades 9170, 9250, and 9370 of SAE Nos. 20, 30, and 40 respectively, are covered by the specification. They may be straight refined mineral oil with or without additive agents.

Detail Requirements – The chemical laboratory test requirements are shown in Table 1.

Underwood Oxidation and Corrosion Test – This test is made on the new oils, without the addition of oxidation catalysts, for 40 hr at 250 F, and at 325 F, on alkali-hardened babbitt, copper-lead and cadmium-silver bearings. Typical results of corrosion tests on additive and non-additive oils are shown in Fig. 1.

General Requirements – The general requirements of the N.S. 9000 series oils and the corresponding procedure for determining compliance are listed in Table 2.

Diesel Engine Tests – The designations, specifications, and operating conditions covering 11 representative diesel engines most frequently used at the Engineering Experiment Station for tests on additive and non-additive type lubricating oils are shown in Table 3. Not all of the types of laboratory or service diesel engines used in oil tests are covered by this paper. However, results from medium and high-speed, four-cycle engines (1000 to 1800 rpm) and medium and high-speed (750 to 1800 rpm) two-cycle engines are presented. Tests on these engines, considered together with the supplementary service trials usually made on diesel engines by the Forces Afloat, are believed to be sufficiently broad in scope to comply with the intent of the specification.

Inspections of Diesel Engines – Complete inspections of the cleanliness of engine parts; measurements of wear of piston rings, cylinder liners, and connecting-rod bearings of all small special test engines; chemical analyses of the used oil samples; and photographs of pertinent engine parts, are made. Included in the engine inspection are the evaluation and recording of the following items:

1. Piston-ring sticking (free, sluggish, pinched, stuck).
2. Piston-ring groove deposits.
3. Piston-ring land deposits.
4. Deposits in oil-ring slots and holes.
5. Deposits on cylinder liners.
6. Deposits on piston tops (crowns) and on inside of pistons.
7. Cylinder-head deposits.
8. Deposits on inlet and exhaust valves.
9. Deposits in energy cells.

[This paper was scheduled for presentation at the 1942 Semi-Annual Meeting of the Society, which was ruled out because of transportation priorities.]

for NAVAL Diesel Engines

by EDWIN N. KLEMGARD
Senior Petroleum Technologist, U. S. Naval Engineering Experiment Station

10. Rocker-arm cover deposits.
11. Rocker-arm shaft deposits.
12. Push-rod chamber deposits.
13. Oil screen deposits.
14. Connecting-rod deposits.
15. Crankshaft deposits.
16. Oil filter deposits.
17. Deposits in bottom of crankcase.
18. Scavenge air port restriction.
19. Crosshead (piston skirt) lacquer or deposits.
20. Condition of connecting-rod bearings.

Tentative Demerit Rating System – Any unfavorable deposit or adverse engine condition attributable to the quality of the oil under test is observed and recorded. Thorough inspections of all the foregoing engine parts are made, but all results are not included in the demerit rating. A simplified form of demerit-rating system has been developed to expedite the reporting of results and to eliminate, in so far as practicable, the personal equation in the inspection of engine parts. The system affords a ready means of comparing and averaging the results of lubricating oil tests. Five types of engine parts are included under engine cleanliness. The overall wear rating includes demerits of the wear of cylinder liners, compression rings, and connecting-rod bearings. A new or perfectly clean surface is assigned a demerit rating of zero. An exceedingly unfavorable condition is denoted by the number "10."

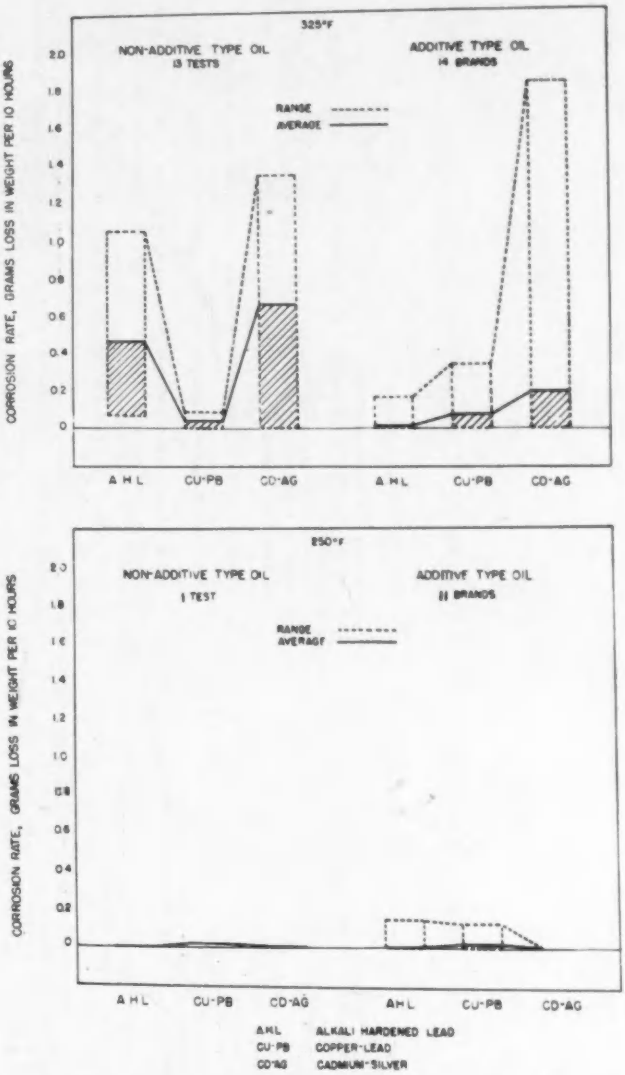
I. Engine Cleanliness Demerits

(1) *Ring Sticking* – The demerit ratings of compression or oil rings are assigned as follows:

Condition of Rings	Demerit
All free	0
One ring sluggish	0.5
One ring 0-75 deg pinched	1
One ring 75-150 deg pinched	2
One ring 150-225 deg pinched	3
One ring 225-300 deg pinched	4
One ring 300-360 deg pinched	5
One ring 0-75 deg stuck	6
One ring 75-150 deg stuck	7
One ring 150-225 deg stuck	8
One ring 225-300 deg stuck	9
One ring 300-360 deg stuck	10

A *free* ring is one that falls in its groove of its own weight when the piston is moved quickly from a vertical to a horizontal position.

A *sluggish* ring is one which will not fall of its own



■ Fig. 1 – Comparative corrosion rates of three bearing alloys with straight mineral and additive-type lubricating oil in the Underwood oxidation apparatus

Table 1 – Detail Requirements – Diesel-Engine Lubricating Oils

Symbol	Grade		
	9170	9250	9370
Viscosity (Saybolt Universal), sec at 130 F	140-200	220-280	320-430
Flash point, (min.), F	350	370	400
Pour point (max.), F	0	10	15
Carbon residue (max.) (ash free), %	0.9	1.1	1.3
Neutralization No. (max.)	0.5	0.5	0.5
Precipitation No. (max.)	None	None	None
Corrosion	None	None	None
Ash (max.), %	0.6	0.6	0.6

Table 2 – General Requirements of N.S. 9000 Series Oils and Corresponding Test Procedures

REQUIREMENT	TEST PROCEDURE
(1) SHALL PROVIDE SATISFACTORY LUBRICATION FOR ALL ENGINE PARTS AND GENERATOR BEARINGS OF NAVAL DIESEL ENGINES.	TESTS IN LABORATORY, SUBMARINE TYPE AND OTHER NAVAL DIESEL ENGINES. TESTS IN NAVY WORK FACTOR MACHINE.
(2) SHALL BE NON-CORROSIVE TO BEARINGS AND ENGINE PARTS.	TESTS IN LABORATORY, SUBMARINE TYPE AND OTHER NAVAL DIESEL ENGINES. UNDERWOOD OXIDATION AND CORROSION TEST. TESTS IN NAVY WORK FACTOR MACHINE.
(3) SHALL NOT CAUSE RING STICKING OR CLOGGING OF OIL CHANNELS.	TESTS IN LABORATORY, SUBMARINE TYPE AND OTHER NAVAL DIESEL ENGINES.
(4) SHALL MAINTAIN MINIMUM PISTON RING AND CYLINDER LINER WEAR.	TESTS IN LABORATORY, SUBMARINE TYPE AND OTHER NAVAL DIESEL ENGINES.
(5) QUALITY SHALL NOT BE ADVERSELY AFFECTED BY MECHANICAL OR FIBEROUS TYPE FILTERS OR BY CENTRIFUGAL PURIFICATION.	TESTS IN LABORATORY, SUBMARINE TYPE AND OTHER NAVAL DIESEL ENGINES, AND SPECIAL FILTER TESTS. CENTRIFUGE TESTS OF NEW OIL WITH AND WITHOUT FRESH AND SEA WATER.
(6) ADDITION OF NEW OIL TO USED DIESEL ENGINE OIL SHALL NOT CAUSE SLUDGING.	MAKE BLENDS OF EQUAL VOLUMES OF NEW AND USED OILS, SHAKE FOR 20 MINUTES, CENTRIFUGE IN TUBE FOR 1 HOUR.
(7) SHALL NOT CAUSE EXCESSIVE CARBON DEPOSITS ON ANY PART OF ENGINE.	TESTS IN LABORATORY, SUBMARINE TYPE AND OTHER NAVAL DIESEL ENGINES.
(8) EACH GRADE SHALL BE SATISFACTORY IN ALL TYPES OF DIESEL ENGINES ORDINARILY REQUIRING THAT GRADE OIL.	TESTS IN LABORATORY, SUBMARINE TYPE AND OTHER NAVAL DIESEL ENGINES.
(9) MIXTURES OF ADDITIVE AND STRAIGHT MINERAL OILS SHALL PERFORM AS WELL AS THE STRAIGHT MINERAL OIL ALONE.	THREE HUNDRED-HOUR LABORATORY DIESEL ENGINE TESTS ON EQUAL VOLUMES OF CONTRACT 3065 AND ADDITIVE OIL
(10) ADDITIVE OIL SHALL NOT BE AFFECTED ADVERSELY BY 2% BY VOLUME OF SEA OR FRESH WATER.	THREE HUNDRED-HOUR, LABORATORY DIESEL ENGINE TEST ON OIL PLUS 2% BY VOLUME OF SYNTHETIC SEA WATER (2% BY VOLUME ADDED EACH 25 HOURS). MEASURE EQUAL VOLUMES (50 ML.) OF NEW OIL AND SYNTHETIC SEA WATER, AND NEW OIL AND DISTILLED WATER IN CENTRIFUGE TUBES; SHAKE FOR 20 MINUTES, CENTRIFUGE IN TUBES, FOR 20 MINUTES. DETERMINE % OF STABLE EMULSION AND LOSS OF ADDITIVE.
(11) ADDITIVE AGENTS SHALL REMAIN UNIFORMLY DISTRIBUTED THROUGHOUT THE OIL AT TEMPERATURES FROM 10°F ABOVE THE POUR POINT UP TO 250°F.	TENTATIVE HOMOGENEITY TESTS: (A) 50 ML. SAMPLE DRIED AT 212 TO 220°F FOR 3 HOURS; COOLED TO 10°F BELOW POUR POINT; OBSERVE AT 55 TO 65°F (B) HEAT 150 ML. SAMPLE AT 250°F FOR 3 HOURS; OBSERVE FOR SEPARATION, THICKENING OR CLOUD FORMATION.
(12) DIESEL ENGINE LUBRICATING OIL SHALL SHOW SUPERIORITY OVER STRAIGHT N.S. MINERAL OILS WHERE RUN IN LABORATORY DIESEL ENGINES FOR AT LEAST 250 HOUR TEST PERIODS AND WHEN USED IN SERVICE.	TESTS IN LABORATORY, SUBMARINE TYPE AND OTHER NAVAL DIESEL ENGINES.
(13) SHALL BE COMPATIBLE WITH ALL OTHER DIESEL ENGINE LUBRICATING OILS PREVIOUSLY PROCURED.	BLEND EQUAL VOLUMES OF NEW OILS (50 ML. OF EACH); SHAKE FOR 20 MINUTES; CENTRIFUGE IN TUBE FOR 1 HOUR.

Table 3 – Specifications and Test Conditions for 11 Diesel Engines Used in Tests of Heavy-Duty Lubricating Oils

DIESEL ENGINE DESIGNATIONS	Atlas-Lanova	General Motors 1-71	Caterpillar 1-Cylinder	Winton X1/201A	Caterpillar D-3400	Winton V12/201A	Cummins H	General Motors 6-71	General Motors 268-A	General Motors 16-248 A	Fairbanks-Morse 38D8-1/8
1. Number of Cylinders	1	1	1	1	4	12	6	6	8	16	9
2. Operating Cycle, Strokes	4	2	4	2	4	2	4	2	2	2	2
3. Bore and Stroke, inches.	3-1/8"x3-3/4"	4-1/4" x 5"	5-3/4" x 8"	8" x 10"	3-3/4" x 5"	8" x 10"	4-7/8" x 6"	4-1/4" x 5"	6-1/2" x 7"	8-3/4"x10-1/2"	8-1/8" x 10"
4. Test bhp (Continuous)	5	15	19.8	75(82.4)	22.8	(a)	95	88(150)	Special Duty Cycle	(b)	(c)
5. Test Speed, rpm	1800	1200	1000	750	1200	450-750	1600	1200(1800)	1300 max.	(b)	(c)
6. B.M.E.P., psi	76.6	70	75.4	78.8	68.1	0-82.9	70	70 (75)	---	---	---
7. Piston Displacement, per.cyl.cu.in	28.7	70.9	207.9	502.2	55.4	502.2	112.1	71.0	166.1	631.4	1037.0
8. Compression Ratio	15.68:1	16:1	---	16:1	---	16:1	17:1	16:1	16:1	15.86:1	14.1:1
9. Connecting Rod Bearings Material	Cadmium-Silver	Copper Lead	Babbitt	Satco	Babbitt	Copper Lead (upper) Satco (lower)	Copper Lead	Copper Lead	Satco	Tri-Metal	Satco
Diameter, in.	2"	2-3/4"	3-1/4"	6"	3"	6"	3-5/32"	2-3/4"	4-1/2"	6-1/4"	6-3/4"
Length, in.	1-1/2"	1-25/32"	2-13/16"	3-7/8"	1-7/8"	6-5/8"	1-30/32"	1-25/32"	2-3/4"	3-1/4"	3-3/4"
10. Fuel Injection Pump, Type	Plunger	Unit Inject.	Plunger	Unit Inject.	Plunger	Unit Inject.	Plunger	Unit Inject.	Unit Inject.	Unit Inject.	Plunger
11. Fuel Nozzle, Type.	Pintle	Multi-Orifice	Single-Orifice	Multi-Orifice	Single-Orifice	Multi-Orifice	Multi-Orifice	Multi-Orifice	Multi-Orifice	Multi-Orifice	Multi-Orifice
12. Crankcase and Sump, Lubricating Oil Capacity, quarts	3-1/2	8	6	60	10	500	28	28	220	1200	1400
13. Crankcase Oil Temperature, °F	135	225	145	150	155	150	170	210(240)	150	155	155
14. Cylinder (Coolant Out) Temperature °F	175	175	175	140	175	150	175	175	150	150	150
15. Duration of Test, Hours	300	300	300 (500)	Variable (250-2500)	300 (500)	250 (500)	250 (300)	300	Variable 250-500	Variable 250-500	Variable 250-500

DAILY SPECIAL DUTY CYCLES SIMULATING SUBMARINE OPERATION,
VARIOUS ENGINE PARTS

REFFRENCE

(a)

TIME	LOAD, hp.	SPEED, rpm
0.5 hrs.	900	750
0.5 "	0	450
0.5 "	282	510
0.5 "	395	570
0.5 "	534	630
0.5 "	701	690
5 min.	810	675
5 "	0	675
5 "	810	675
5 "	0	675
5 "	810	675
5 "	0	675
11.5 hrs.	720	675
8.0 "	900	750
0.5 "	945	750
23.5 hrs.		

(b)

TIME	LOAD, hp.	SPEED, rpm
0.5 hrs.	2000	800
0.5 "	0	400
0.5 "	488	500
0.5 "	733	575
0.5 "	1075	650
0.5 "	1480	725
5 min.	1800	720
5 "	0	720
5 "	1800	720
5 "	0	720
5 "	1800	720
5 "	0	720
11.5 hrs.	1600	720
8.0 "	2000	800
0.5 "	2100	800
23.5 hrs.		

(c)

TIME	LOAD, hp	SPEED, rpm
0.5 hrs.	2000	900
0.5 "	0	450
0.5 "	343	500
0.5 "	592	600
0.5 "	941	700
0.5 "	1405	800
5 min.	1800	810
5 "	0	810
5 "	1800	810
5 "	0	810
5 "	1800	810
5 "	0	810
11.5 hrs.	1600	810
8.0 "	2000	900
0.5 "	2100	900
23.5 hrs.		

weight when the piston is moved quickly from a vertical to a horizontal position, but can be moved by moderate finger pressure.

A *pinched* ring is one that does not move in its groove under moderate finger pressure, but has a bright or polished face over its entire circumference, showing that it was essentially free during operation.

A *stuck* ring is one that does not move under moderate finger pressure and, in addition, its face is covered by lacquer or carbon over any part of its circumference, showing that the ring was stuck during engine operation.

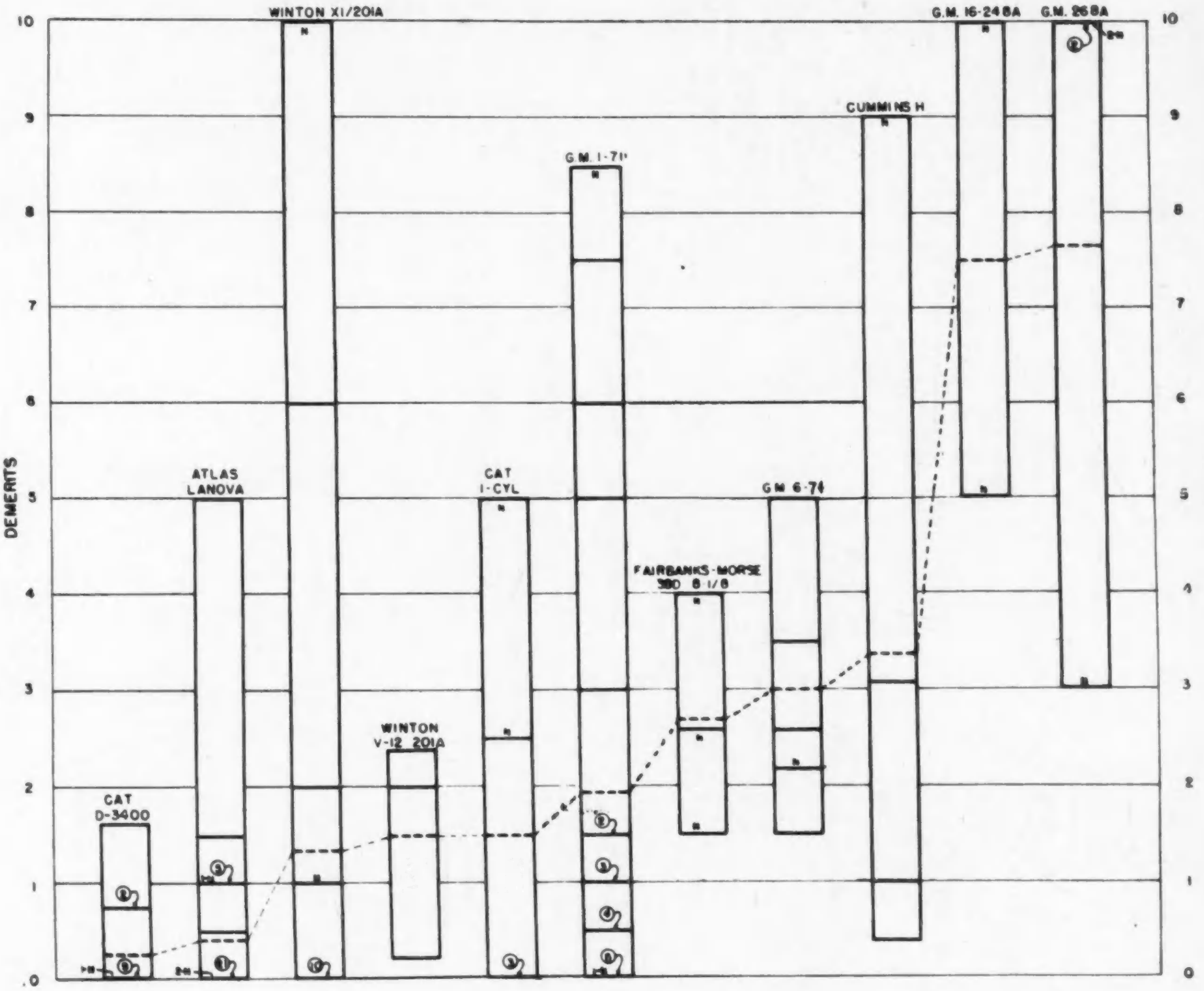
The demerit rating of a single piston shall be the numerical sum of the individual ring-sticking demerits, and shall have a maximum value of 10. For multicylinder engines, the ring-sticking demerit shall be the numerical average of the individual ratings of each piston.

(2) *Valve Sticking* – One completely stuck valve, causing the engine to stop, shall be rated 10. Inspection showing no sluggish action of the valve in its guide, shall be recorded as having a zero demerit rating.

(3) *Liner Lacquer* – When all of the area of the liner which is swept by the piston rings is definitely discolored by blowby gas or coated with resins, varnish, carbon or other oil-decomposition products that are not easily removed by wiping with a clean cloth, it shall have a demerit rating of 10. For a multicylinder engine, the rating shall be the average of the ratings of all liners.

(4) *Crosshead Lacquer* – Pistons having all of the skirt area below the upper oil rings coated with carbon, lacquer, or other oil-decomposition products not readily removed by wiping with a cloth, shall have a demerit rating of 10. For a multicylinder engine, the rating shall be the average of the ratings of all pistons.

(5) For two-cycle engines, an additional demerit shall be included in the Overall Cleanliness Rating, namely: the *Scavenge Air Port Restriction* demerit. For perfectly clean air ports, the rating shall be zero; for each 5% of the total air port area restricted, the demerit rating is 1. For 50% restriction, the demerit shall be 10.



Average demerits of all oils tested in each engine are connected by a broken line.

Each horizontal solid line represents one lubricating oil test except where the figures in circles show the number of oil tests having the same demerit. All oils tested were of additive type except where the designation "N" indicates a non-additive type oil.

■ Fig. 2 – Ring-sticking demerits for 11 diesel engines

(6) *Overall Engine Cleanliness Rating* – This is the average of the four or five foregoing demerit ratings.

II. Wear Demerits

(1) *Cylinder-Liner Wear* – A rating of 10 shall be assigned when the average wear of a liner (transverse and longitudinal) at the upper end of travel of the top compression rings on one piston becomes 0.004 in./1000 hr/in. of liner diameter. No measurable wear, as compared with the original dimensions of the cylinder liner, shall have a demerit rating of zero. The demerit rating for a multicylinder engine shall be the average ratings of all cylinder liners.

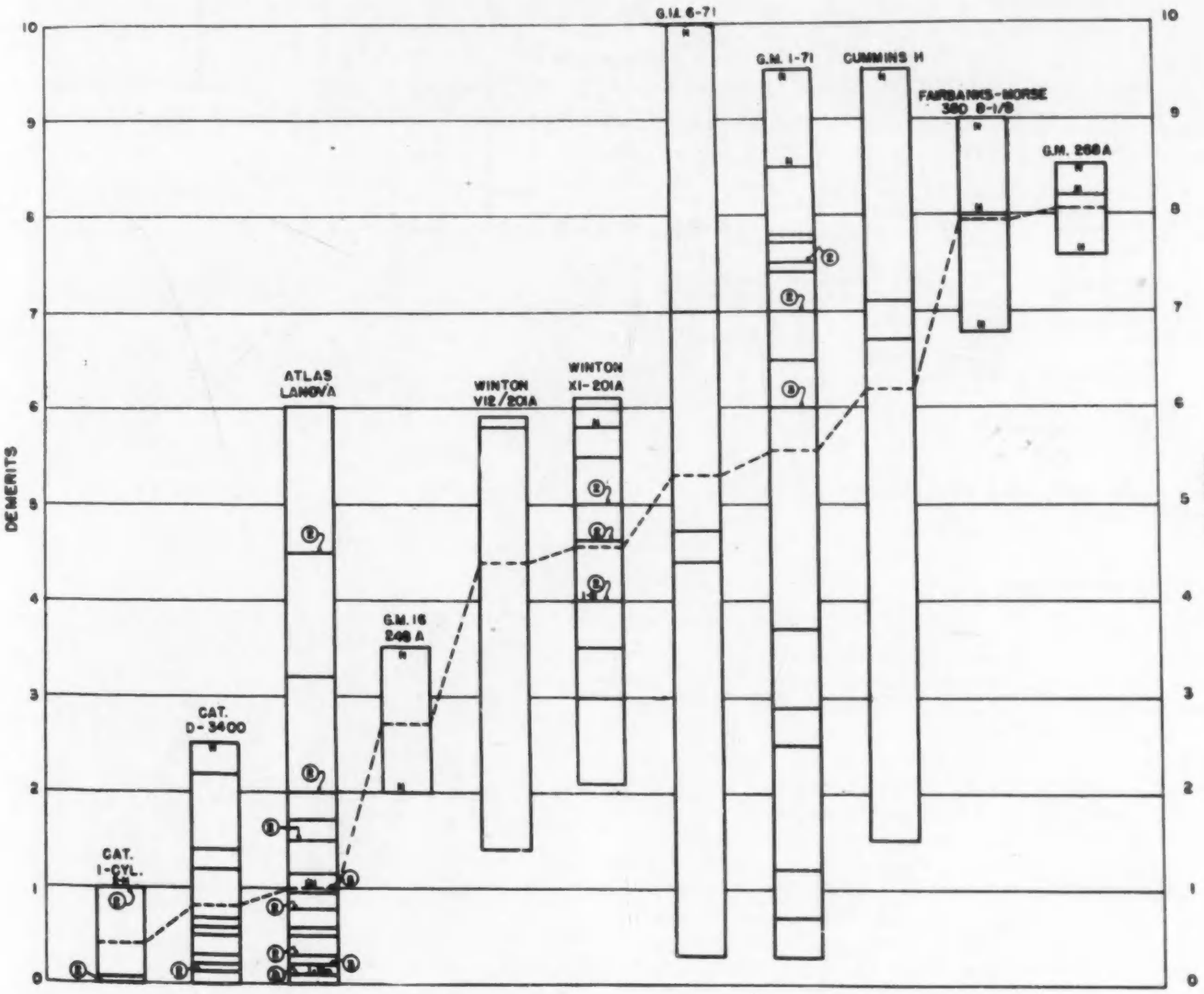
(2) *Compression Ring Wear* – A demerit rating of 10 shall be given when the per cent loss in weight of compression ring(s), in the top groove becomes 10% of its (or their) original weight, computed on the basis of 1000 hr of operation. The compression-ring wear demerit rating for a multicylinder engine shall be the average ratings of the top rings of all pistons. For no measurable loss of weight, the demerit rating shall be zero.

(3) *Bearing Wear and Corrosion* – A demerit rating of 10 shall be assigned when the loss of weight of a connecting-rod half-bearing shell becomes 0.10 g/sq in. of projected bearing area, computed on the basis of 1000 hr of operation. Only the single half-shell having the maximum loss of weight for a single or multicylinder engine shall be reported. For no measurable loss of weight, the demerit rating shall be zero.

(4) *Overall Wear Rating* – This is the average of the three foregoing demerit ratings.

III. Oil Stability Demerits

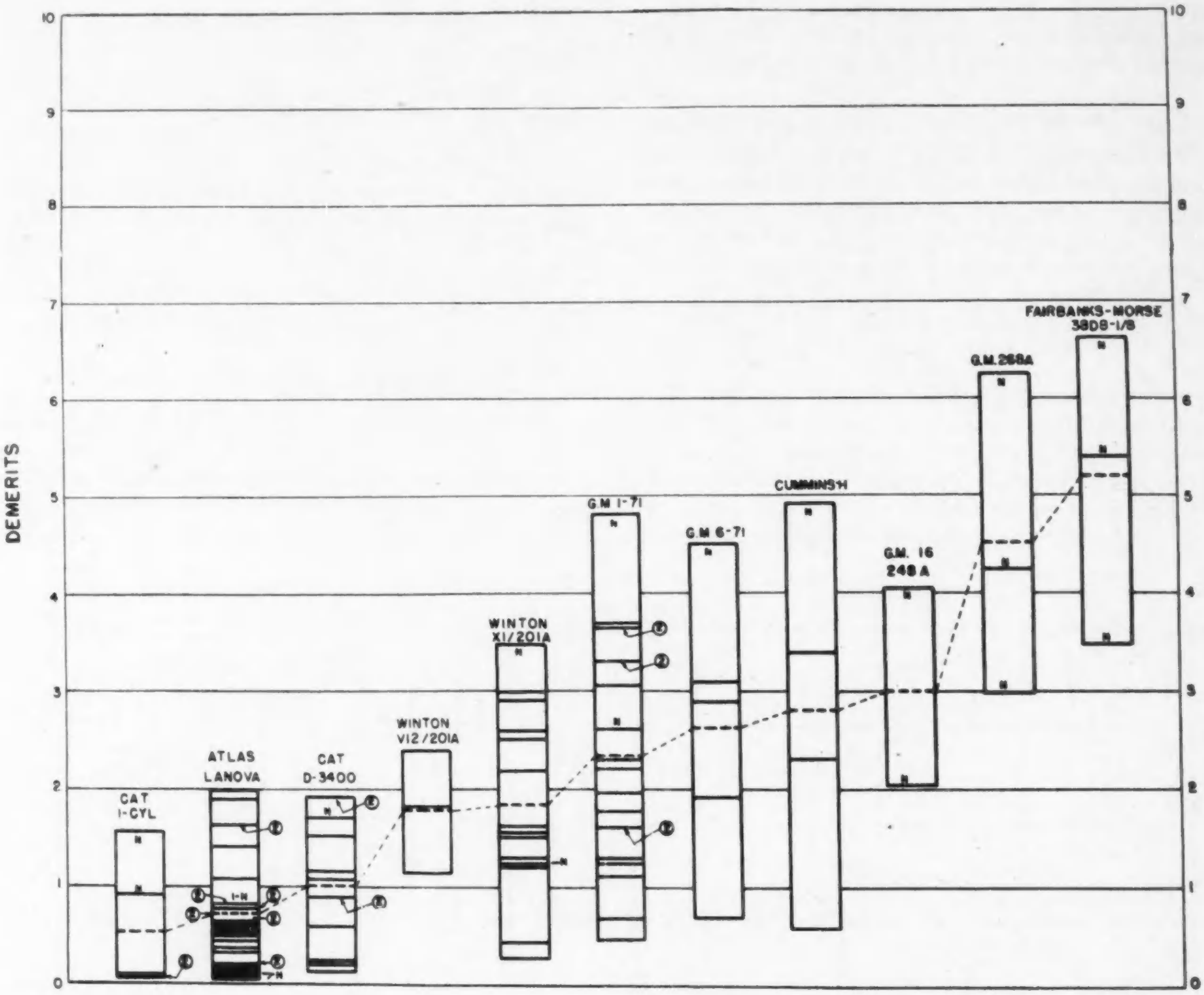
Satisfactory correlation of the results of chemical laboratory tests (precipitation number, naphtha insolubles, and chloroform solubles) with engine cleanliness results or wear has not been possible with the data available. This result is probably due to the wide variety of oils and



Average demerits of all oils tested in each engine are connected by a broken line.

Each horizontal solid line represents one lubricating oil test except where the figures in circles show the number of oil tests having the same demerit. All oils tested were of additive type except where the designation "N" indicates a non-additive type oil.

■ Fig. 3 – Crosshead lacquer demerit ratings for 11 diesel engines



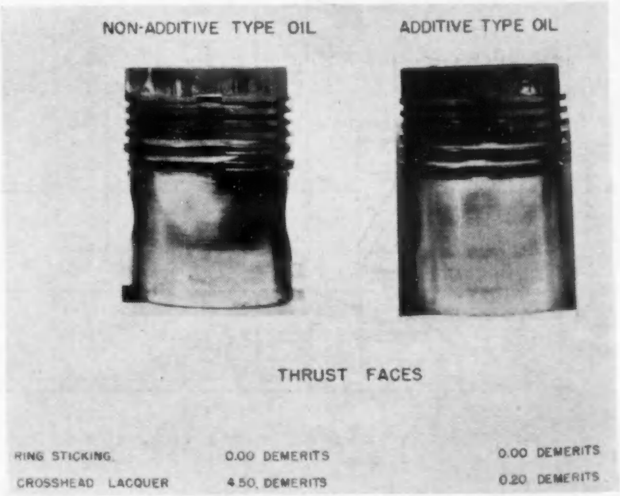
Average demerits of all oils tested in each engine are connected by a broken line.

Each horizontal solid line represents one lubricating oil test except where the figures in circles show the number of oil tests having the same demerit. All oils tested were of additive type except where the designation "N" indicates a non-additive type oil.

■ Fig. 4 – Overall engine cleanliness demerits for 11 diesel engines

engines used. Oil stability demerits based on the usual increases in Saybolt viscosity, precipitation number, per cent naphtha insolubles and per cent chloroform solubles, show changes in and contamination of used oils. The increases in these test results are, however, unreliable as a general means of predicting such practical and important engine conditions as ring sticking and wear of cylinder liners, piston rings, and bearings. Consequently there is a persistent need for improved methods for analyzing and testing used oil samples. Laboratory tests showing more significance regarding pertinent changes in oils and the effects of oils on engines are desired in the development of a rational oil stability demerit system. Typical tests desired are:

(1) Wear or scratching due to contaminants in used oil.
(2) Corrosivity of used oil.
(3) Sludge or lacquer deposition.
(4) Additive deterioration or consumption.
(5) Detergency.



■ Fig. 6 – Pistons from Caterpillar D-3400 diesel test engine after 300-hr tests on non-additive and additive type oils

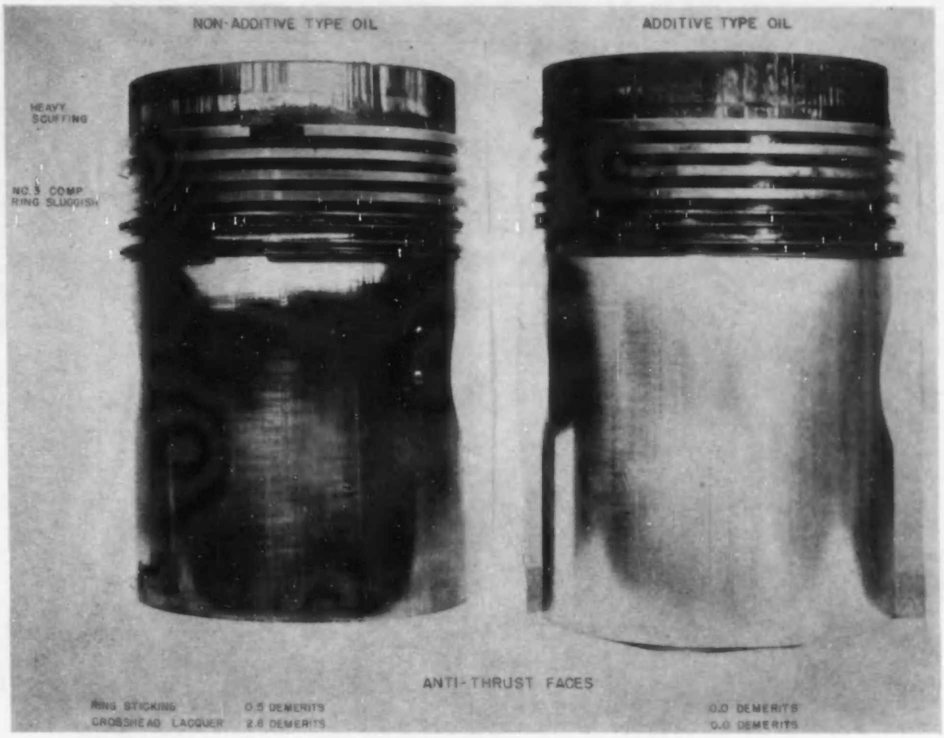


Fig. 5 – Pistons from Caterpillar 1-cyl diesel test engine after 500-hr tests of non-additive and additive type lubricating oils

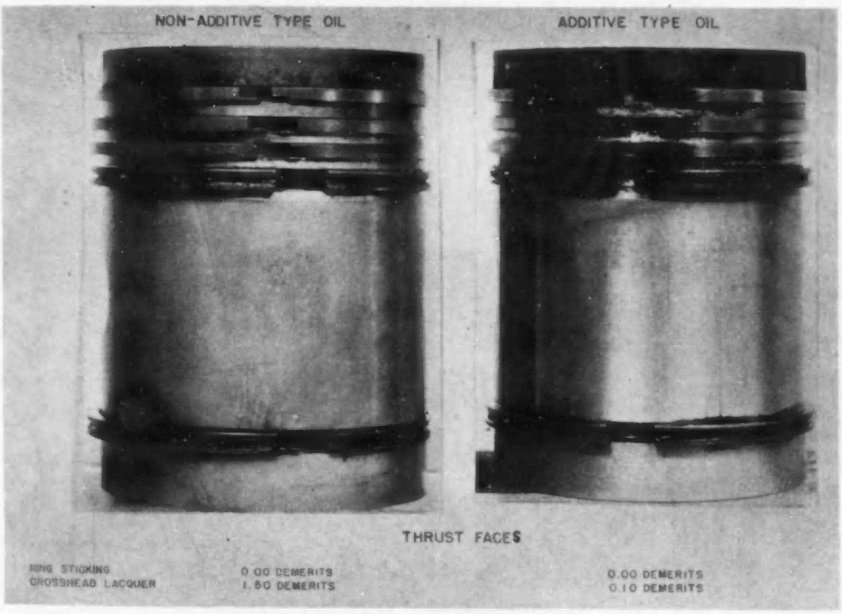
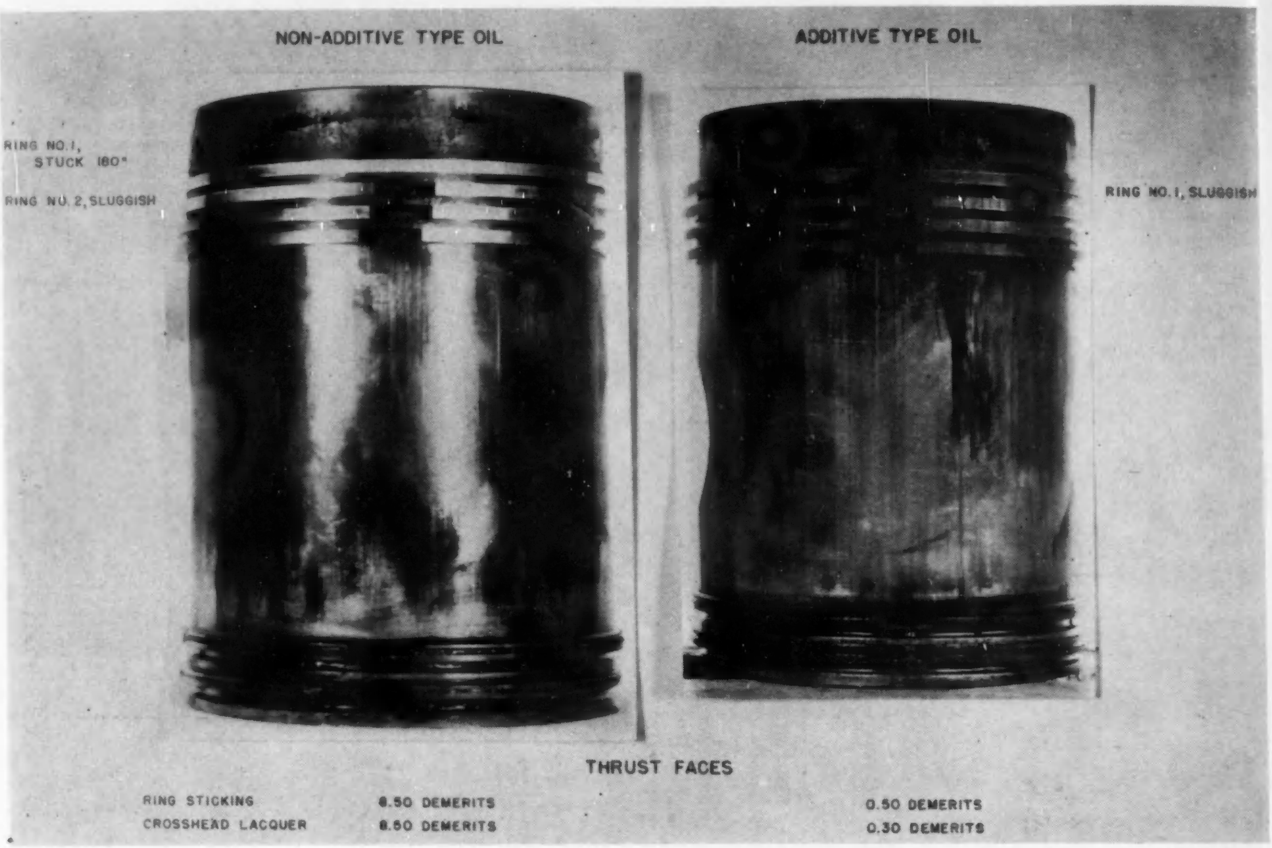


Fig. 7 – Pistons from Atlas-Lanova 1-cyl diesel test engines after 300-hr tests of non-additive and additive type lubricating oils

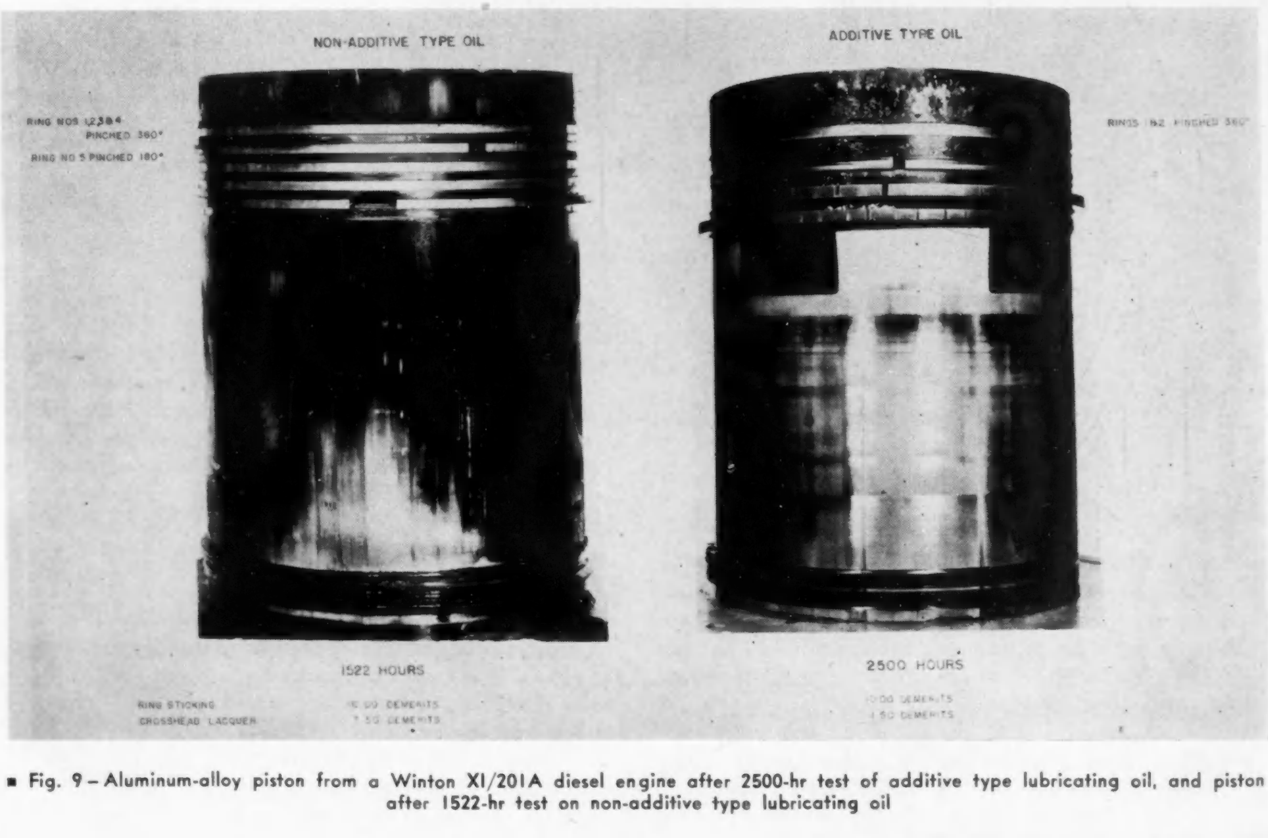
Severity of Tests

Piston-ring sticking, lacquering, and bearing corrosion are related to the time that the oil is subjected to high temperatures in the engine, as well as to operating conditions, design of pertinent parts, and other items. In tests made in diesel engines at the Engineering Experiment Station, time is stressed by making tests of 250 to 2500 hr duration without changing the oil under test, only sufficient make-up oil being added as required to maintain practical engine operating conditions. Temperatures of coolant and lubricating oil are maintained as shown in Table 3.

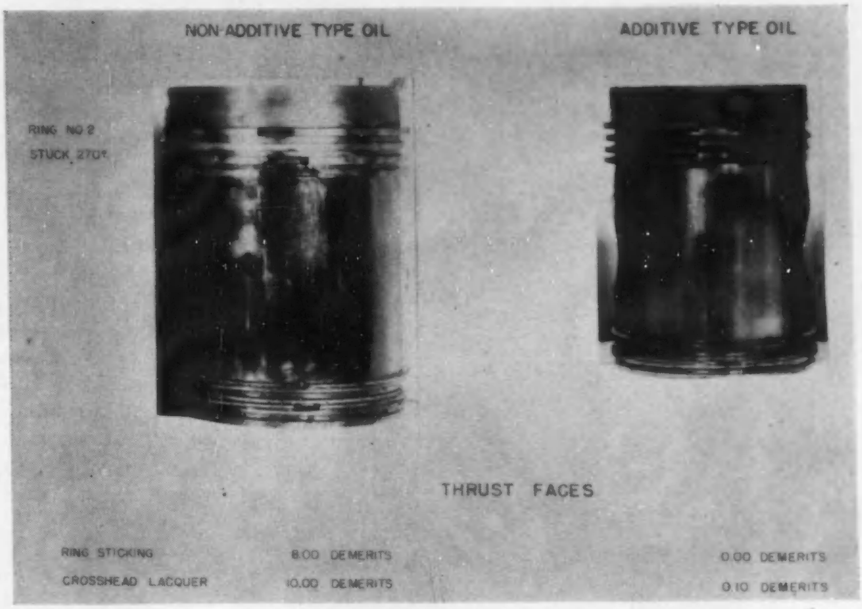
Rate of Oil Deterioration – It is realized that the rate of deterioration of the lubricating oil, subjected to temperatures of 400 to 550 F in the ring zone, for instance, will be a great many times higher than in a sump maintained at 150 F. It is therefore possible to pinch or stick rings in some diesel engines operating with the oil in the crankcase at low temperatures (150 F) and showing relatively little deterioration. The temperatures in the ring zone and the tendency to stick rings is dependent on piston and ring



■ Fig. 8 – Pistons from GM 1-71 1-cyl diesel test engines after 300-hr tests of non-additive and additive type lubricating oils



■ Fig. 9 – Aluminum-alloy piston from a Winton XI/201A diesel engine after 2500-hr test of additive type lubricating oil, and piston after 1522-hr test on non-additive type lubricating oil



■ Fig. 11 – Cast-iron pistons from GM 6-71 diesel engines after 300-hr tests of non-additive and additive type lubricating oils

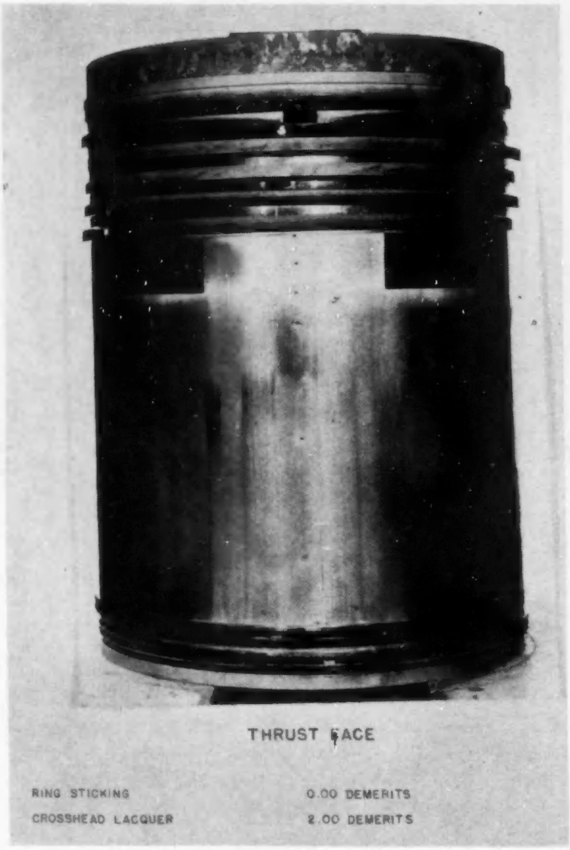
■ Fig. 10 (below) – Cast-iron pistons from Cummins H diesel engine after 250-hr tests on non-additive and additive type lubricating oils

design, wear, clearances, deposits, metallurgy, and operating conditions.

Ring Sticking in Service – A few cases of ring sticking have been reported in diesel engines in service, operating on highly detergent oils. Piston assembly design and severity of operating conditions were probably responsible for such cases of ring sticking. The effective detergency of the oils in the ring zones was insufficient to prevent ring sticking under existing operating conditions. Additives will decrease, but not totally prevent, ring sticking in some diesel engines. A combination of the best designs and oils of most universally favorable characteristics is the desired objective.

Operating Conditions – The maximal operating condi-

■ Fig. 12 – Cast-aluminum-alloy piston with bevel-type compression rings from Winton XI/-201A diesel engine after 446-hr non-stop reliability test at 750 rpm and 82.45 bhp on non-additive type lubricating oil



■ Fig. 13 – Aluminum-alloy piston with bevel-type compression rings from Winton V12/201A S/M Type diesel engine after 500-hr test of additive type lubricating oil, on special-duty cycle

tions are based on military requirements placed on boats and ships to fulfill successfully their war missions.

New Designs – The development of improved piston designs, piston rings, and liners for submarine-type engines is believed to be of primary importance. The data presented, which cover lubricating-oil tests in submarine-type or other large marine diesel engines, should be interpreted with the understanding that some of these engines were experimental. For this reason the results on different oils tested in large diesel engines may not be directly comparable, but are intended to show a trend dependent on the requirements imposed by new designs and prescribed operating conditions.

■ Discussion of Results

(1) *Piston-Ring Sticking* – Fig. 2 shows the range in piston ring-sticking demerit ratings of additive and non-additive type lubricating oils tested with few exceptions for 300 hr, without change of oil, in 11 types of diesel engines. The extent of the improvement in ring sticking attributed to additive type oils is shown by the bar graphs. The ring-sticking demerit ratings attained a maximum of 10 demerits in three of the tests. In these three cases, non-additive type oil was under test.

The data show that ring sticking and ring pinching occur in both large and small diesel engines. The ring-sticking demerits ranged from 0 to 10 in the Winton X1/201A diesel engine, due to the effects of various piston assembly designs and oils tested. The minimum average ring-sticking demerit was 0.26 for the Caterpillar D-3400 engine, and the maximum average ring-sticking demerit was 7.7 for the GM 268-A diesel engine.

Need for Improved Heavy-Duty Oils – Heavy-duty oils having improved detergency as related to ring sticking, which will keep the piston-ring grooves free of adhesive, hard, carbonaceous material causing sluggish, pinched, or stuck piston rings, are needed. Research and development work leading to improved base stocks and superior types of additives which are universally effective in all types of diesel engines, should not be relaxed.

(2) *Piston Crosshead Lacquering* – Fig. 3 shows the range in piston crosshead lacquer demerit ratings of additive and non-additive type lubricating oils tested in 11 types of diesel engines. The general improvement caused by additive-type oils is shown in the bar-graphs. Average results show that crosshead lacquering was a minimum (0.41 demerits) in the Caterpillar 1-cyl test engine, and a maximum (8.1 demerits) in the GM 268-A diesel engine, with intermediate values for nine other engines.

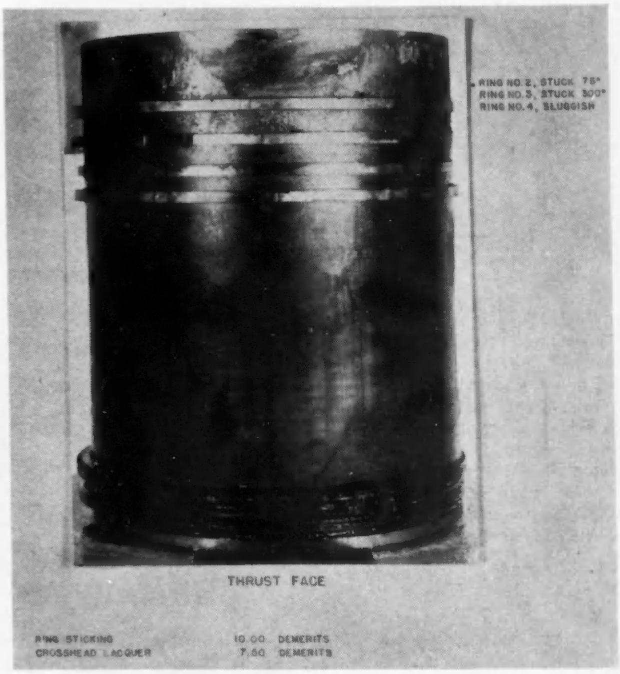
Detergent Properties – The demerit ratings show the necessity for improving the detergent properties of lubricating oils, particularly those used in submarine-type diesel engines. Small areas of thin lacquer on piston skirts are probably not harmful, but large areas of thick, gummy or hard lacquer (Fig. 12) which reduce heat transfer from the piston, increase friction and rub off and collect in piston-ring grooves, are objectionable. An effective oil for preventing piston lacquering should have:

(a) High detergency (dispersion of contaminants and oil degradation material, by polar bodies in the oil).

(b) High solvency for gums and resins.

(c) High thermal stability.

(d) The property of forming soft rather than hard de-

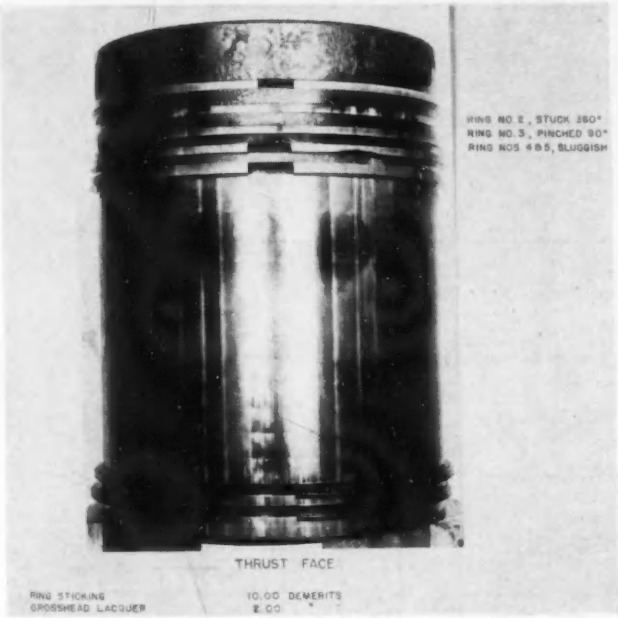


■ Fig. 14 – Standard cast-iron piston from GM 8-268-A diesel engine after 240-hr test of non-additive type lubricating oil, on special-duty cycle

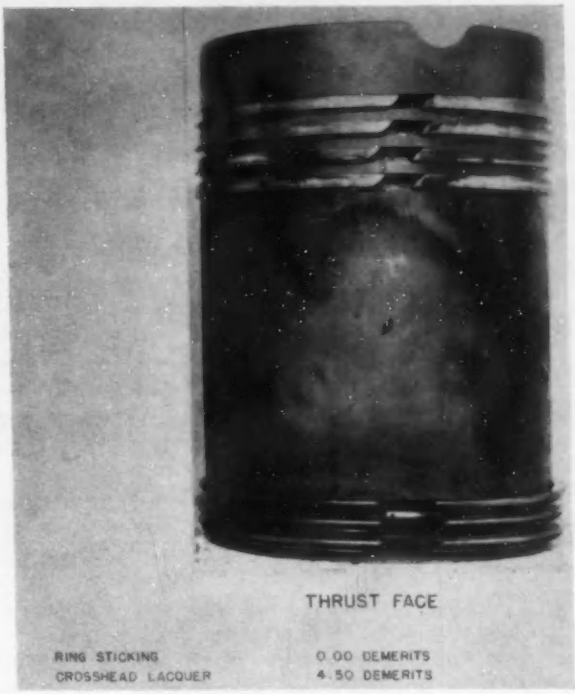
posits where conditions are severe enough to produce any solid adherent material.

(3) *Overall Engine Cleanliness* – Fig. 4 shows the range in and averages of the overall engine cleanliness demerits for 11 diesel engines. All of the ratings of 3 demerits or more are for large engines, the GM 16/248-A, GM 268-A and Fairbanks-Morse 38D8-1/8 diesel engines.

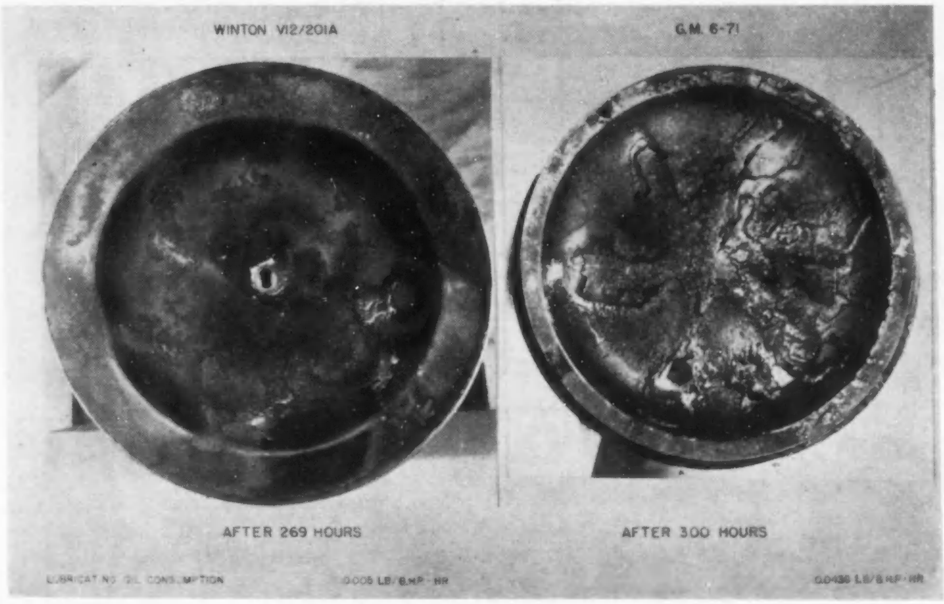
Caterpillar Diesel Engine Tests – The Caterpillar 1-cyl diesel engines had the lowest average overall cleanliness rating, 0.52, as compared with the next lowest, 0.71 for the Atlas-Lanova engines. The Caterpillar 1-cyl engine is believed by many engineers to require oils having a high degree of detergency. Fig. 5 shows that an additive oil causes better piston cleanliness than non-additive oil in this engine. As shown by the data in Fig. 4, Caterpillar 1-cyl diesel engines were cleaner when lubricated with detergent oils than the other 10 diesel engines used in these tests. The Caterpillar D-3400 diesel engine was effective in showing improvements attributable to detergent-type additives, Fig. 6. However, the average overall cleanliness ratings obtained in 12 tests of 300-hr duration was 1.00 which is less than the corresponding averages for 8 other types of diesel engines.



■ Fig. 15 – Standard cast-iron piston from GM 16-248-A S/M Type diesel engine after 250-hr test of non-additive type lubricating oil, on special-duty cycle



■ Fig. 16 – Standard cast-iron piston from Fairbanks-Morse 9-cyl, opposed-piston, model 38D8-1/8 S/M Type diesel engine after 1686-hr test of non-additive type lubricating oil, on special-duty cycle (511 hr since oil change)



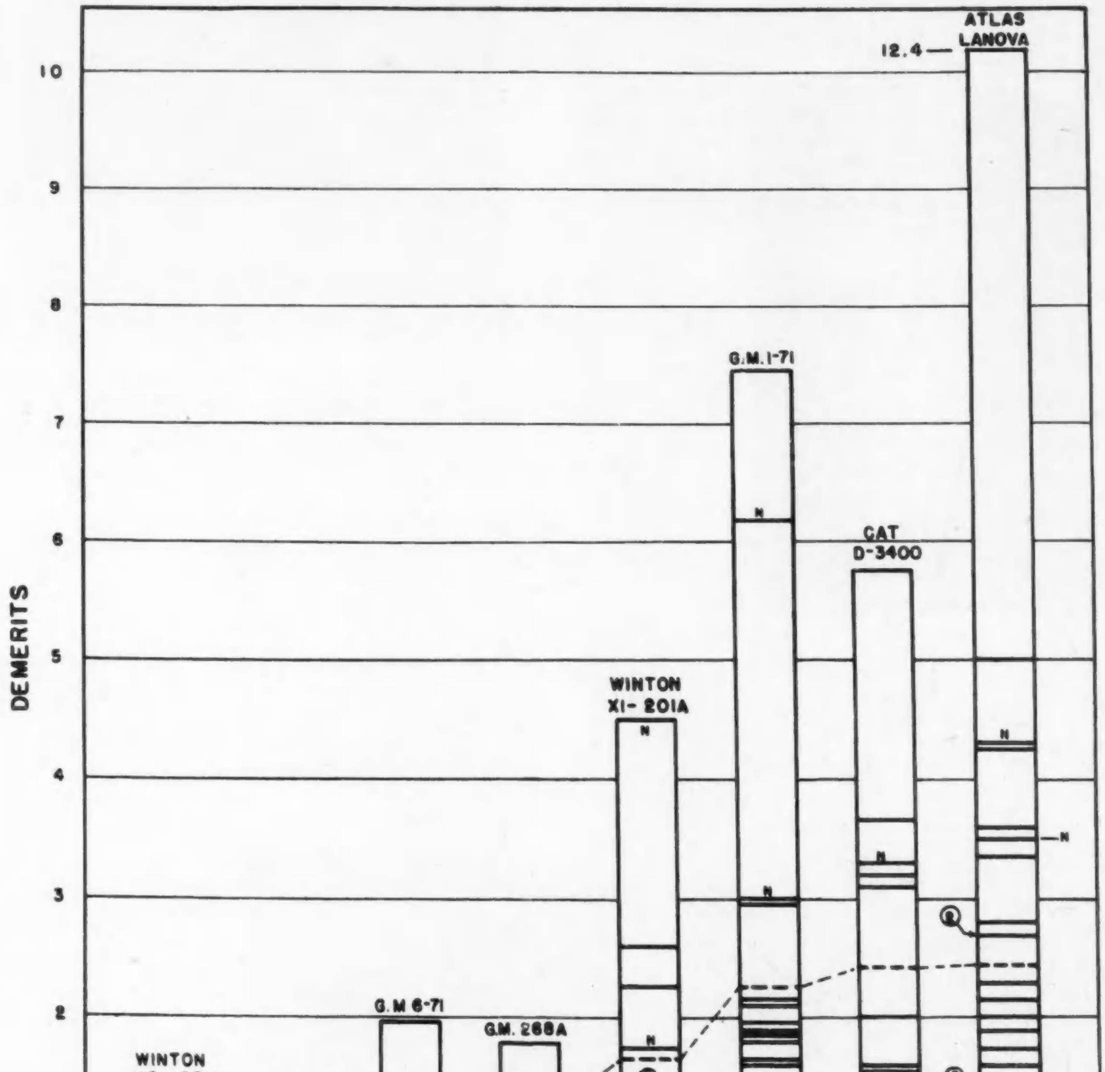
■ Fig. 17 – Tops of pistons from Winton V12/201A and GM 6-71RC diesel engines after tests on additive-type lubricating oils, showing nature of deposits formed

■

Improvements Attributable to Additive-Type Lubricating Oils – Typical pistons, after tests on non-additive and additive type lubricating oils, from Caterpillar 1-cyl, Caterpillar D-3400, Atlas-Lanova, GM 1-71, Winton X1/201A, Cummins H, and GM 6-71 diesel engines are shown in Figs. 5, 6, 7, 8, 9, 10, and 11 respectively. In each case the piston lubricated with additive type oil is superior to the piston lubricated with non-additive type oil, in either ring sticking, crosshead lacquer, or both, as shown by the demerit ratings.

The Atlas-Lanova engine does not stress the detergent characteristics of oils, as shown by the pistons in Fig. 7, and the 0.71 average overall cleanliness demerit rating for 29 oils tested, Fig. 4.

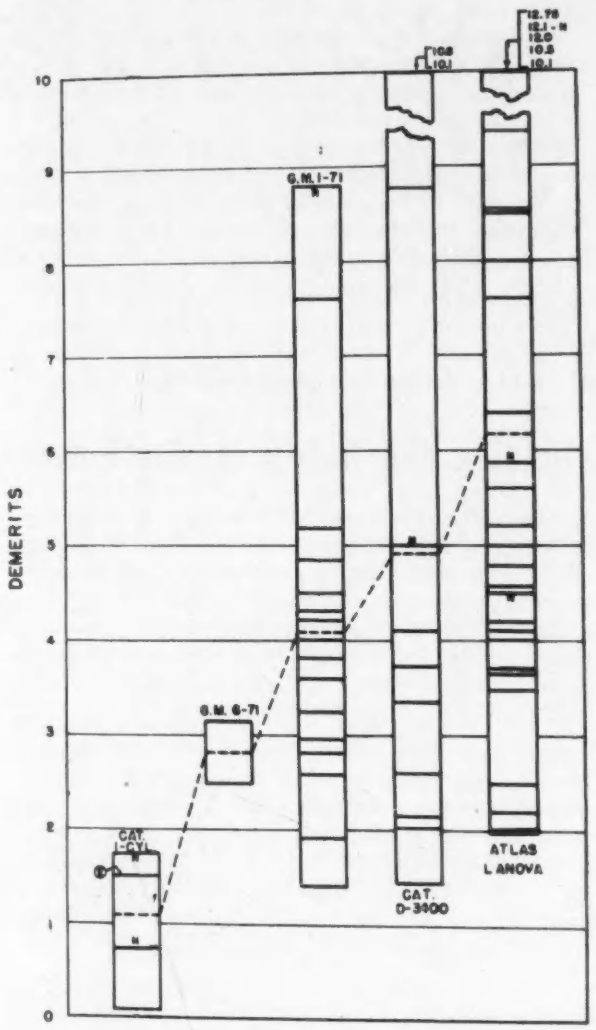
The tests made in GM 1-71 diesel engines tend to show much greater differences in the detergent nature of the oils, Fig. 8, than do those made in Atlas-Lanova engines. The average overall engine cleanliness demerit rating of all oils tested in GM 1-71 engines was 2.33 as compared with 0.71 for Atlas-Lanova engines, Fig. 4. The average overall cleanliness demerit based on five 300-hr tests in the GM 6-71 diesel engine was 2.62. The average engine



Average demerits of all oils tested in each engine are connected by a broken line.

Each horizontal solid line represents one lubricating oil test except where the figures in circles show the number of oil tests having the same demerit. All oils tested were of additive type except where the designation "N" indicates a non-additive type oil.

■ **Fig. 18 – Cylinder-liner wear demerits for eight diesel engines**



Average demerits of all oils tested in each engine are connected by a broken line.
Each horizontal solid line represents one lubricating oil test except where the figures in circles show the number of oil tests having the same demerit. All oils tested were of additive type except where the designation "N" indicates a non-additive type oil.

■ **Fig. 19 – Piston-ring wear for five diesel engines**

cleanliness results obtained in GM 1-71 and 6-71 diesel are of the same order. See Figs. 2, 3, and 4.

Winton Diesel Engines – Fig. 12 shows unfavorable piston lacquering and ring sticking in a 446-hr test of non-additive type oil in a Winton X1/201A diesel engine equipped with an experimental piston submitted for test. Fig. 9 shows unfavorable ring-sticking and piston-lacquering in a 1522-hr test of non-additive type oil with a different type of experimental piston. Fig. 9 also shows a piston used in a 2500-hr test of an additive-type oil. The oil was not changed during the 2500-hr test. Two pinched compression rings were noted in the 250-hr inspection and, after 2500 hr, they had not become stuck.

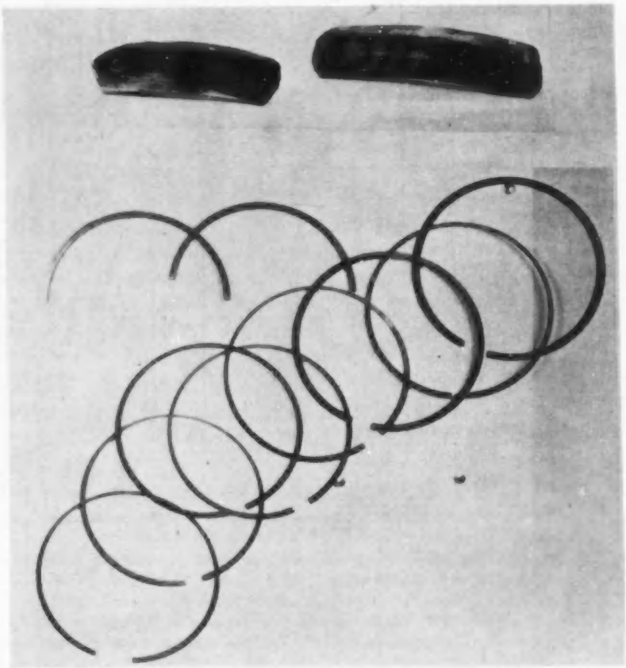
The additive-type oil caused an increase in cleanliness of the Winton X1/201A piston over non-additive-type oil, but there was evidence that further improvements in the detergent property of additive-type oils for this type of engine were desirable. Fig. 13 shows favorable results on an experimental-type piston from a Winton V12/201A after a 500-hr test of a satisfactory additive-type oil.

Large Diesel Engines – Unfavorable piston cleanliness results obtained in tests of non-additive type oil in GM 8-268-A and GM 16-248-A diesel engines are shown in Figs. 14 and 15, respectively. Fig. 16 shows the irregularly nicked and worn condition of the ends of the compression rings of a piston from the Fairbanks-Morse 38D8-1/8 diesel engine. This condition was not attributable to the quality of the non-additive type oil used. High temperatures in the ring zone have permitted the formation of only a moderate deposit of lacquer and carbon.

Ash Residues – Fig. 17 shows unfavorable ash residues on the tops of pistons, which resulted from the use of certain additive type oils in Winton V12/201A and GM 6-71 diesel engines. Additive-type oils which do not cause these deposits, although oil consumption may be high due to worn cylinder liners and piston rings or other causes, are considered desirable.

(4) *Cylinder-Liner Wear* – Fig. 18 shows cylinder-liner wear demerits based on inches increase in diameter per 1000-hr per in. of liner diameter for eight types of diesel engines lubricated with additive and non-additive type oils. In general, wear of cylinders has been greater in the small diesel engines than in the submarine-type diesel engines. In the Atlas-Lanova 5-bhp engine, the average cylinder wear was 2.43, the highest average for eight diesel engines. Low wear was reported for the Winton V12/201A and Caterpillar 1-cyl diesel engines. There have been few cases of excessive cylinder wear. At least one of the additive oils was superior to non-additive type oil in cylinder-liner wear in seven types of diesel test engines.

(5) *Piston-Ring Wear* – Fig. 19 shows higher average piston-ring wear for Atlas-Lanova diesel engines than for



■ **Fig. 20 – Piston rings from S/M Type diesel engine showing excessive wear in less than 500 hr of operation on additive-type oil**

the other four engines for which data were available. Excessive wear due to stuck piston rings from submarine-type diesel engines is shown in Fig. 20. One or more of the additive oils was superior to non-additive type oil in piston-ring wear in five types of diesel test engines.

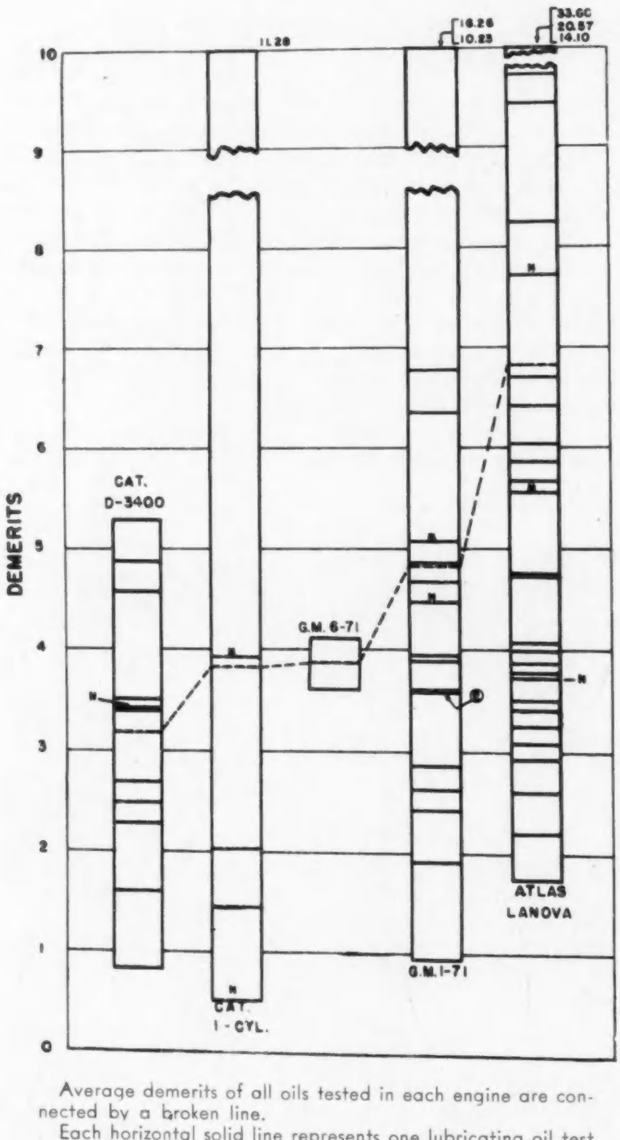
(6) *Overall Engine Wear* – Fig. 21 shows the average overall wear demerits, including wear demerits of cylinder liners, piston rings and connecting-rod bearings. Atlas-Lanova diesel engines had the highest average overall wear demerit rating. One or more of the additive oils was superior to non-additive type oil in overall engine wear in four types of diesel engines.

■ Conclusions

1. Heavy-duty type lubricating oils containing metallo-organic compounds were effective in decreasing, but not totally preventing, piston-ring pinching and sticking in Naval diesel engines.

2. In general, one or another of the additive-type lubricating oils was superior to non-additive type oil in piston-lacquering, piston-ring sticking, and wear of liners and piston rings in the test engines.

3. Research and development work leading to improved base stocks and superior types of additives which are universally effective in all types of diesel engines should be continued. A combination of the best diesel engineering designs and heavy-duty lubricating oils of most universally favorable characteristics, is considered to be the desired objective.



Average demerits of all oils tested in each engine are connected by a broken line.
Each horizontal solid line represents one lubricating oil test except where the figures in circles show the number of oil tests having the same demerit. All oils tested were of additive type except where the designation "N" indicates a non-additive type oil.

■ **Fig. 21 – Overall engine wear demerits for five diesel engines**

Winning the Battle of Maintenance

THIS war has been defined as a "War of Production," but it is also a battle of maintenance, for what good is all this equipment that we produce if we fail to maintain it, but permit it to stall on the repair line for lack of lubrication, "stitch-in-time" adjustments, and major repairs? Surely there is a job to be done of teaching men to drive trucks so as to conserve them, and teaching others to repair and maintain this equipment.

Right now, to the best of my knowledge, the Army needs trained mechanics in order to secure that quality of maintenance which will affect the "stitch-in-time" adjustments and repairs, and will build back quickly into the equipment operating miles equivalent to the original mileage – and not half or a third of it. Trained mechanics are no better than the caliber of their training or the instructors and materials available to train them.

It seems to me that quality of maintenance is something which initially must come from the automotive industry. Who can better define rebuilding practices than those who originally built the product?

Maintenance is definitely a phase of the automotive industry, and improvement of maintenance methods and the general caliber of maintenance is very much an obligation of the industry, rather than a problem for operators to struggle with individually. Operators – whether they be the Army or of civilian status – are fully aware of their obligations regarding maintenance and its effects on the life of their equipment but, when we consider methods and how quality of maintenance is to be secured, then I think we should look to the manufacturer to take the initiative.

There is a tremendous educational job to be done in which the automotive industry should take the lead, and which should be designed to reach every Army mechanic. By its very nature it would also reach the mechanics of branches and dealers, and likewise reach the large group of mechanics found in the shops, large and small, of fleet operators.

Excerpts from the paper: "The Battle of Maintenance in the War of Production," by J. Willard Lord, safety engineer, The Atlantic Refining Co., presented at a Philadelphia Section Meeting of the Society, Philadelphia, Pa., March 11, 1942.

Effect of War Development on POST-WAR CAR DESIGN

by FRANK JARDINE

Chief Engineer, Castings Division, Aluminum Co. of America

CONDITIONS governing post-war car design are changing so rapidly that anything written on this subject the first part of the year would be out of date by the end of the first quarter. The attitude toward work on post-war cars has changed; also, the policy of the car manufacturers and their ideas regarding conditions under which a post-war car will be brought out. If the war lasts a long time, conditions will be different than if it ended quickly. If it suddenly comes to an end, the design of the first post-war car would be different than it would be if the end were gradual and long drawn out. It is not intended that this paper should have anything to do with the question of when or how the war will end, even though the post-war car design will be closely connected with these questions.

Automobile production was discontinued so suddenly that manufacturers and dealers not only had a great many finished cars on their hands, but there was and still is a large inventory of parts on hand which must be used in new cars to be built in the future, as service parts for cars already built and sold, or melted up as scrap. Since production was discontinued, the machinery for manufacturing cars has been dismantled and replaced by war production machinery in practically all automobile manufacturing plants. Before a new car can be built, the war material equipment will have to be removed and replaced with automobile production equipment which will take some time after war production is discontinued.

Before it is decided what kind of a post-war car to

[This paper was scheduled for presentation at the 1942 Semi-Annual Meeting of the Society, which was ruled because of transportation priorities.]

THE post-war car will have about the same general appearance as the pre-war car, but it will be better streamlined, more efficient, and lighter. Although it will be smaller, it will be long enough to ride well and wide enough to carry three people in the front seat. Performance will be almost as good as that of present-day cars, and gasoline economy will be better. Price will be lower than that of the 1942 models.

Ha Ha

The foregoing is a brief composite picture of the post-war car as envisioned by a majority of passenger-car engineers queried by Mr. Jardine. These opinions were sent to the author in answer to a questionnaire sent out to a number of engineers who will have most to do with design of the post-war car. Excerpts from a number of these opinions are quoted in this paper.

Discussing the effect of the length of the war and of inventories, Mr. Jardine opines that, if the war comes to a sudden end, the first post-war car brought out will be the same as the 1942 models from a design standpoint, to be followed by the real post-war car after sufficient time has elapsed really to develop a new car.

With an estimated production of aluminum 6½ times that of 1939 by the end of 1943, and with a steadily falling price trend of this metal, he sees an ever-increasing amount of aluminum used in passenger cars, much of it in parts never before made of aluminum in production. By the use of aluminum and with careful design, he contends, the weight of the average car can be reduced 1000 lb without reducing the car size noticeably. More magnesium will be used, he believes, for the same reasons. Plastics also will be used in increasing amounts, he predicts, but he does not look for applications other than trim and small parts for some time.

With reference to fuel, he reports the general belief that 80-octane gasoline will be the regular grade; 100-octane gasoline, the premium grade. The price will be up due to taxes.

Although the rear-engine car is receiving more attention than ever before, Mr. Jardine reports that there are only a few engineers who believe that some post-war cars will be built with this engine arrangement. The frameless chassis, he believes, will find new applications in post-war cars. Hydraulic drives, with either fluid flywheels or torque converters, it is predicted, will come into increasing use.

THE AUTHOR: FRANK JARDINE (M '16) has spent 24 years in the aluminum branch of the automotive industry, first with the Aluminum Castings Co., then with Aluminum Manufacturers, Inc., and now with Aluminum Co. of America. As chief engineer of the Aluminum company's Castings Division, he is in charge of all engineering connected with automobile and aircraft castings – sand, permanent-mold, and die – and forgings. Prior to entering the aluminum branch of the industry, he worked with a number of pioneer automobile companies.

build, somebody must decide what to do with the parts inventory on hand; also, whether or not the old machine tools used for pre-war cars will be set up and the parts already manufactured used, or a completely new car manufactured. If the old car is put back into production, until a new car can be developed by some manufacturers, there are likely to be other manufacturers who will disregard completely the last model built and bring out an entirely new model if they have time. The public will no doubt be most interested in the new model, and the manufacturer bringing out a new model will not only attract the public, but the dealers also. It takes time to bring out new models and, until this war is won, nobody will have time for new designs, unless it comes to a slow, indefinite finish, allowing time to reorganize the automobile engineering departments so a new design can be ready when the war is over. The most common opinion indicates that the war will come to a sudden end and, under those conditions, the first post-war cars brought out will be the same as the 1942 models from a design standpoint, regardless of what they are called. The real post-war car will not appear until enough time has elapsed really to develop a new car. In the meantime, the public will use the same kind of a car they have been using during the past years. The first post-war car will be high-priced.

Ever since the automobile became a standard product, it has been considered efficient manufacturing to use as many parts from previous models as possible; also, as many previous-model tools and fixtures, for any new models brought out. If the first post-war car built is to be the same as the 1942 car, there will be practically no change in this practice after the war and the post-war car will start where the pre-war car left off, except that the designers will have the advantage of war-time developments, and also, of other war-time conditions. Considerable experience has been gained substituting one material for another, due to shortages, and it will be possible to take advantage of lower prices of material made in larger quantities than ever before, such as aluminum, magnesium, and plastics, also of developments in materials, products, and processes during the war.

There will be a different grade of gasoline and oil available. There is very little automobile development work going on at this time which could be used for post-war designs. Aircraft design development does not always apply to cars. It also will be necessary for the designer to consider the cost of tires and taxes both on the car itself and on gasoline, as the cost of operation of new cars will have to be in line with our revised standard of living and economic situation.

■ Important Materials Changes

The aircraft developments adaptable to automobiles will not be outstanding in post-war car design, but there will be real changes in materials. By the end of 1943 the U. S. production of aluminum will be approximately 2,100,000,000 lb which is 6½ times the 1939 production. Each of several of the new plants being built will produce more aluminum than the entire nation made at the World War I peak. Increased production has brought about a price reduction from 20¢ per lb to 15¢ per lb within the last two years. This increase in production, also the decrease in price, should result in a large increase in the amount of aluminum used. The present shortage of primary aluminum has resulted in a large-scale substitution of secondary aluminum alloys for primary aluminum alloys which will mean a greater use of secondary alloys than ever before. The cost of these secondary alloys should be lower than ever before which would mean cylinder heads, crankcase, and cylinder blocks and transmission cases at prices never before possible. The entire aluminum industry has increased its capacity far beyond anything ever considered in the past, and built modern production plants for fabricated parts. Practically all the foreign countries producing aluminum before the war have increased their capacities in proportion to the increases in the aluminum production of the U. S., and the available supply of aluminum in most any form has been increased many times. The amount of magnesium available after the war will be increased in practically the same proportion as aluminum, making this material available in large quantities. The fabricating technique as applied to both these materials has been greatly improved, making not only better but cheaper parts of these materials possible for post-war production.

In order to keep pace with the demands for plastics as a substitute for other materials, the plastic industry has expanded greatly both in fabrication and production of its many different materials. These materials can be obtained in many different forms and will be available for new uses in the post-war car.

Many people insist that plastics will be used in ever-increasing quantities in automobiles, but to date all indications point to its use as trim and for small parts. Other uses will have to be developed after the war.

The ferrous metal industry is also making many changes. A new crankshaft foundry has been built and put into production on war work which will no doubt make cast crankshafts for the general automobile trade after the war.

While rubber is scarce at this time, it is being used in a great many new parts and new synthetic rubber plants are being built to increase the supply of this material.

Higher octane gasoline and better oil also will make possible new improvements and better performance. The price will be up due to taxes, requiring more efficient engines to hold the cost of operation down.

In order to obtain as many opinions as possible on the real post-war car from people who will have the most to do with the design of this car, questionnaires were sent out to these people and their opinions tabulated. After going over these questionnaires, it is possible to describe a car which fits most of the opinions fairly closely.

There seems to be some difference of opinion regarding the appearance of the post-war car. Most engineers insist that it will look about like our present cars with a little more tendency to streamlining, that it will be smaller, but have no shorter wheelbase, have the same tread and width of seat as at present, and be slightly lower. Other engineers think that we cannot build lower cars on account of high curbs, but a flat floor without the power tunnel would be all that is possible.

■ Opinions of Some Engineers

"In general, I would not anticipate any marked changes for the first year or so after the end of the war. Past experience has shown that the first few years after a war have been those of vigorous if false prosperity and thus should be similar economically to the year or two preceding the war. This factor plus a desire on the part of automobile manufacturers to get their products on the market quickly without the delay necessary to extensive redesigning and retooling

will, I believe, result in no radical difference between post-war and pre-war cars.

"It is probably inevitable, of course, that the war will be followed sooner or later by a depression and that such a depression will encourage the development of smaller, lighter, more economical cars."

Another engineer's opinion on appearance is as follows:

"It is my opinion that any new car brought out of American manufacture will not dare to be a very radical departure from the present-day automobile so far as appearance is concerned. Any change in car appearance will have to be made gradually, as the buying public has never accepted wholeheartedly any radical change in car appearance. I do believe we can expect a further step in the direction of eliminating fenders, as such, and of course running boards are gone. It seems to me there will be a turn against an excessive amount of bright-finish parts such as were hung around this last year's car until it looked like a piece of costume jewelry. The economic situation will force a limitation of some of the unnecessary trimmings. The car will be lower in appearance and, no doubt, more streamlined.

"Post-war cars will be lower, providing that we find a different location for the powerplant. The height of the present-day cars is limited within a certain range by the fact that we have a propeller shaft and transmission under the floor. There is a certain limitation regarding the height of the car as it affects ingress and egress; also, the fact that, if a car is built too low, it is impossible to open a door when parked alongside of present-day curbs in the majority of our cities."

Another opinion on wheelbase:

"I doubt if wheelbase will be much shorter than on the present car. Wheelbase in certain types of design does not necessarily affect cost. An extremely short wheelbase car never has been, and I do not believe ever will be, made to ride as well as a long-wheelbase car. What I am trying to make clear is that an automobile of around 100-in. wheelbase does not ride as well as one of 112 or 114 in., and the difference in cost, provided that the proper type of construction is used, within this range, is negligible."

Another opinion on visibility:

"The stylists have been claiming greater visibility for the last five years, and all they have succeeded in doing is to place the driver where he can't see his own front fenders to keep them from scraping other people's cars, and to expose him to more sun glare in front by sloping the windshields and flattening the rear windows so they won't shed snow or rain. It depends on what you call 'visibility.' "

Another opinion on the size of the car:

"We believe that there will be a trend toward lighter cars which, however, will have as much or more passenger space than do present cars."

There seems to be agreement that the wide seats will be retained, but there is some question as to the location of the passengers as compared with the present cars, as is indicated by the following opinion:

"This will depend entirely upon the powerplant arrangement. Today's automobile (the average car, I am talking about) has the passengers moved about as far forward as is practical and still maintain traction, under certain conditions, for the rear wheels. Some people seem to have the idea that, if a powerplant should be placed in the rear, the passengers can be moved much farther forward. This is not true to any great extent, except in possibly some cars using 8-cyl engines. The degree to which passengers can be moved forward is limited by the space required to turn the front wheels in steering."

The engine will be located in front in most cars, but some engineers think there may be a few rear-engine cars. Nobody seems to be interested in a front-wheel drive with engine in the front or rear. If the rear-engine car were built, the individual rear-wheel springing would be used.

Most engineers favor the valve-in-head motor for future use, slightly smaller than used at present and more economical, as mentioned in the following opinion:

"Conditions after the war may be so different than what we now imagine they will be, that answers to questions concerning engine dimensions or material could be nothing except guesses. For example, the use of aluminum or iron will depend to a great extent upon the relative prices of these materials which we believe nobody is in a position to prophesy. I feel there will be a general trend toward smaller, more economical engines and that radical innovations, such as air-cooled engines, superchargers, and the like will be uncommon, at least for a year or so following the end of the war."

An opinion on superchargers is as follows:

"If high-octane fuels are to be used, I believe that their use will be associated with supercharging since supercharging really pays dividends with high octane numbers. If superchargers are used, it is possible that the centrifugal type will be satisfactory if used in conjunction with an automatic transmission. The centrifugal supercharger, of course, is of no use in increasing engine torque at low speeds unless a complicated supercharger drive is used. However, with the automatic transmission, high engine torque at low engine speed is never required."

An opinion on frames is as follows:

"I believe that frames will not disappear suddenly since a more or less complete elimination of a frame introduces serious manufacturing problems. There probably will be an evolution toward the frameless car but, during the first few years of this evolution, we believe that partial frames will be retained. (By partial frame I mean sufficient frame to hold the chassis together during assembly.)"

Other engineers feel that the conventional frame will pass out in the near future, while some are still in favor of the conventional frame.

The question of transmission and clutch would be taken care of by most engineers as follows:

"Hydraulic drive will become universal either through the medium of a fluid torque converter with probably two automatic gear speeds or a fluid coupling with multiple automatic gear positions."

Another opinion is:

"Fluid flywheels are here to stay in some form or other. The post-war automobile should bring out some type of torque converter which has many advantages that cannot be obtained with the fluid flywheel and the automatic transmission, such as has been produced to date."

Another opinion is:

"Lowest-cost cars – dry disc clutch and manual shift-transmission. Higher-cost cars – torque converter and automatic geared shift."

On the question of plastics one engineer claims:

"Plastics will be used in increased quantities as we progress. Many of our interior decorations which to date have been metal will become plastics – window molding, interior door moldings, instrument panel parts, et cetera. Plastics will predominate."

There seems to be a general opinion that plastics will be cheaper than metal parts used at present, but this is not true in all cases and, while plastics have a very definite field, they also have limitations which make it impractical to use them in a great many places.

A number of engineers claim that crankshaft material will be changed as indicated by the following opinion:

"Motors will be redesigned to use ferrous alloy crankshafts and alloyed cast iron camshafts."

Other engineers claim that crankshafts will be made of SAE 1045 steel, air-quenched for high machinability and Tocco hardened and a cast chrome cylinder iron Tocco-hardened camshaft. General opinion indicates that hardened crankshafts will be used.

The general opinion on aluminum to be used can be summed up in the statement that a large amount will be used for many parts which have never been made of aluminum in production before, provided the price permits.

An opinion on aircooled engines:

"I do not believe that the aircooled engine will find an extensive place unless we go to rear-engine cars. The aircooled engine seems to be inherently noisier than the liquid-cooled engine, and I believe that it will only be tolerable to the American public if installed in a rear-engine car. Furthermore, aircooled engines are a decidedly more serious problem as regards lubricants and I believe any wise manufacturer would hesitate to duplicate the practices of the old Franklin Co. in specifying special lubricants with limited distribution."

Most engineers agree that tire sizes will remain about the same with the possibility of having fewer sizes. There is no very definite opinion as to the kind of rubber.

It is generally believed that our regular grade of gasoline will be about 80 octane with higher octane fuel avail-

able in premium grades. One engineer expresses the following opinion:

"By and large, motor gasoline after the war should be quite similar in quality to that being produced currently, particularly as none of the manufacturing changes, including aviation gasoline production, are to any great extent applicable to motor gasoline. Probably the most outstanding improvements in petroleum products will show up in lubricants. After another two or three years, oils of the so-called heavy-duty type should be generally available in the United States."

Most engineers are of the opinion that the first post-war car will be practically the same as the pre-war cars but, after the first one is in production, it will be necessary to bring out a new car which is different and better to build up public demand for the new and better models as is indicated by the following opinion:

"There is a tremendous inventory of pre-war car parts and material, including tools. This may govern the post-war car design but it is to be hoped that the car manufacturers can see their way clear to step into much of the new design that has been desirable for years but which could not be put into effect because of competition and commercial situations that would not permit scrapping everything for a new start. Post-war production will give the first opportunity in years for a real forward move in car design."

■ Result of Answers to Questionnaires

After looking over all the questionnaires, the following description of the second post-war car would fit the ideas of most engineers. The first post-war car is to be about the same as the standard 1942 cars.

In a general way, this car should look very much like our present cars with more streamlining; it will be smaller, lighter, and more economical. There will be no radical departure from the present lines.

These specifications are inconsistent when compared with the statements that the wheelbase must be no shorter to insure good riding and the seats no narrower to insure seating comfort for the American public. These cars can be made shorter overall without changing the wheelbase or riding comfort. If the American public must ride in comfort, the new cars will be a combination of light materials, careful design, motor efficiency, and a minimum reduction in size.

In order to reduce the weight of a car, it can be made smaller and of the same materials, the same size out of the light materials, or redesigned using the same material as originally used, but with greater economy in the use of material and greater care in design. The post-war car will be designed very carefully and economically, using as much light-weight material as practical, and very careful consideration will be given the size to obtain the lightest car possible without reducing the size to a point where it will be considered too small. By use of aluminum the average car can be reduced 500 lb in weight without going to extremes. By careful design and taking advantage of this possible weight saving, this car should be reduced another 500 lb in weight, making a weight reduction of 1000 lb, without reducing the size of the car noticeably. This weight saving should result in real economies in the way of gasoline consumption, tire mileage, and taxes.

The bumpers will be smaller, lighter, and less prominent. There will be less chromium, less emphasis on the radiator grill. More rear wheels will be covered and, in some cases, there will be an attempt to cover the front wheels. There will be no more long front fenders extending over the front door. Some models may have a clear plastic windshield curved at the corner posts for greater visibility, but clear plastics are not too clear and this design will be very extreme. The fenders will be made to look a part of the body, giving the car a narrower appearance. Further attempts will be made to lower the car. Any increase in the size of the power tunnel in the floor or lowering the roof and the door which is already low enough to give trouble with high curbs, will be considered bad practice.

There will be very few changes in the general appearance and design of the body. Most cars will have steel bodies, but aluminum is being considered for some models, depending on the price. Aluminum doors, rear decks, and hoods are also being considered for cars with steel bodies, as these parts can be made using the same dies as are used for steel parts. The weight saving obtained by using aluminum for these parts not only helps reduce the overall weight of the car, but lighter doors are easier on hinges, have less sag, and are easier to handle. Light hoods and rear deck doors are easier to handle and can be used without counter-balancing springs. Some consideration will be given to plastic body parts and possibly doors, but not for early post-war production.

Some designers will try plastic windows sealed in flush with the outside of the car for better streamlining and lighter, less expensive construction. Sealed-in windows would require either air conditioning or filtered air. Double plastic windows could be used with an air space between to eliminate defrosting troubles.

The headlights will be about the same as at present – in some cars, concealed. The hood will be a little shorter and the passengers moved slightly forward.

The inside trim will be colored plastic or colored aluminum to match upholstery, and it is possible that a new plastic upholstery material may be tried. The instrument lighting and arrangement, also, the cowl lighting, will be improved.

A few engineers recommend a shorter wheelbase, a narrower seat, with smaller engines and less performance. This is more consistent with the small-income, small-car idea which seems to be under consideration by a few manufacturers.

The location of the engine, front or rear, has received a great deal of attention in the past and, at this time, the rear-engine car is receiving more consideration than ever before, but there are only a few engineers who claim there will be some models built with the engine in the rear. This idea seems to need a good sponsor.

There is a real increase in interest in the frameless car or the car with the frame and body combined. In order to reduce weight the steel frame with an aluminum body was suggested. There will, no doubt, be more cars in production with frameless chassis.

Conventional coil springs in front and leaf springs in the rear, or coil springs all around, will be used with little consideration given to the torsion bar, rubber or pneumatic suspension.

Tires will be the same as they are today for size and design, made of either natural rubber or synthetic rubber, depending on which is available at the lowest price.

The steering gear will not be changed from the present conventional type.

Independently sprung front wheels and no front axle will be used more. Stamped steel wheels or stamped aluminum wheels with aluminum rims are considered.

Conventional rear axles will be used on most cars with more attention given to independent rear wheels with conventional drive and lighter axles.

The conventional clutch will be used only on the lowest-priced cars. The fluid flywheel, torque converter or other hydraulic drive with full automatic or semi-automatic transmission is most favored. The old conventional transmission to be used on low-priced cars. The use of a torque converter with one first gear and reverse would reduce the weight of the transmission and clutch assembly and make the use of aluminum die castings possible.

The overdrive is still very popular and will be used in more models. Some models will have it incorporated in the automatic transmission.

Outlook for Engines

While there is some interest in the sleeve-valve engine, there is no indication any will be put in production. There is very little interest in the aircooled engine.

The engine to be used in the post-war cars will be practically the same as the engine used in pre-war cars. The size may be slightly reduced as a result of the improved efficiency that is expected, also of the reduction in weight. Most engineers anticipate no attempt to improve acceleration for post-war cars and some engineers feel that the public will have to accept less acceleration. In order to take advantage of the higher octane fuel and obtain greater efficiency, the valve-in-head engine will be used more, and the trend in engine size will be toward smaller bores and the most economical bore-stroke ratio. The tax situation will tend to make smaller engines more attractive.

Water cooling will be used for practically all engines with practically no change in water pumps or circulation system. Some work will be done on other forms of liquid cooling to obtain higher operating temperatures and higher efficiency.

The interest in aluminum engines is increasing, but this interest seems to be based entirely on price. The aluminum engine would include a semi-permanent mold cylinder block and crankcase, cylinder head, flywheel housings, and other complicated aluminum castings. Most of these parts should be of secondary alloy. The permanent mold and semi-permanent mold process is more adaptable to high-speed economic operation which results in lower prices. Gears will be used for camshaft drive as an economy.

The overhead-valve type of aluminum engine will use either the wet sleeve or dry sleeve, but the L-head aluminum engine should use the dry sleeve. The wet sleeve is much more economical to make in the foundry. Automatic valve-lash adjusters should be used on aluminum engines.

Furnace brazed permanent-mold cast-aluminum intake manifolds with polished inside passages will be used.

There will be a real increase in the use of aluminum cylinder heads, and the aluminum piston will be practically standard. The conventional carburetor will be used, but some consideration will be given fuel injection. The supercharger is still of some interest, but its use will depend largely on the development of a satisfactory drive. There is practically no interest in anything but the conventional solid poppet valves, except where the sleeve-valve engine is being considered. Valve-lash adjusters will be used on more engines to insure quiet valves at all times. The hardened crankshaft, also the cast crankshaft, will be used.

Improved steel-backed bearings will be used; also, aluminum-alloy bearings on hardened shafts.

The conventional pressure lubrication systems will be retained in practically all cars with minor changes in pumps and methods of distribution.

Rubber engine mountings are accepted as standard, but there will be a marked improvement in these mounts for future cars.

Considerable work has been done in an effort to substitute other metals for copper in the radiators, but copper will be used in most radiators with some consideration given to aluminum and steel.

More plastics will be used, more aluminum, and some magnesium, but all these materials will be competitive with other materials.

Hydraulic brakes will be used in most cars with very little consideration given to other types of brakes.

The weight of all cars will be greatly reduced. The medium-priced cars will weigh approximately 2500 lb, and other price-class cars will be reduced in weight in the same ratio.

The cost of tires will be increased. The cost of gasoline will also be increased if not by the cost of manufacture, by the amount the gasoline tax is increased. The total taxes on a new car will be increased, but income will be decreased, so the price of cars must be decreased.

The price of a car in the Ford, Chevrolet, Plymouth class will be $700, and the most expensive car approximately $2000. The ideas on price were not consistent, as the small-car price varies from $500 to $1000 and the large-car price from $1200 to $3000.

The $700 cars should travel 30 mpg of gasoline and use 80-octane fuel, and the $2000 car should travel 20 mpg of gasoline and use 100-octane fuel.

Summary of Questionnaires

When summarizing all the questionnaires received, most engineers agree that the post-war car should have about the same appearance as the pre-war cars, or no radical change in appearance, be better streamlined, smaller, more efficient, and lighter, but it must be long enough to ride well, wide enough to carry three people in the front seat, and have almost as good performance as the present-day cars and with better gasoline consumption.

Some engineers claim that the post-war cars should have about the same appearance as pre-war cars, be better streamlined, smaller, more efficient and lighter with shorter wheelbase, slightly narrowed seats and less performance, but with much better gasoline consumption.

Both groups of engineers agree that the price of the post-war car must be lower than 1942 car prices.

It would appear to be possible for both groups to reach the same goal, but the first group will have a much more difficult job. They will have to use all the light-weight material possible, such as aluminum bodies, hoods, engines and wherever else possible; also, magnesium and plastics. Their steel and iron parts would have to be designed very carefully to insure minimum weight, and the motor would have to be most efficient and well engineered.

The problem for the second group would not be so difficult because it would be easier to reduce weight by reducing the size of the car, but the smaller car would not be as acceptable to the public.

I wish to take this opportunity to thank the many men in the automotive industry who contributed to this paper by answering the letters and questionnaires sent out by the writer.

THE POSITIVE-DISPLACEMENT S

by JOHN L. RYDE
Chief Engineer, McCulloch Engineering Co.

SINCE the operation of the positive-displacement supercharger is characterized by a fixed displacement of air per revolution and delivery is affected only to a small amount by pressure variations, the supercharger continues to deliver extra air to the engine at the lower speeds. For applications requiring a high torque at the lower speeds, this characteristic is advantageous; it imparts the desired "lugging" ability to the engine in contrast to centrifugal units where the pressure, being proportional to the square of the rpm, falls off rapidly as the speed is reduced.

In spite of the fact that the theoretical efficiency of the Roots type of positive-displacement supercharger is not as high as that of the type which compresses the air within the blower before delivery, the Roots type has the simplest design, having but two moving parts; requires no lubrication in the blower air chambers, the only points of friction being the bearings, seals, and gears.

This paper presents a preliminary means of estimating the performance of a positive-displacement supercharged engine and offers a yardstick of the practical limitations for the performance of the units with present-day equipment. It is shown in a discussion of valve overlap, intercooling, and scavenging that the combined effect of the intercooler and the use of scavenging permits the highest increase in output of any supercharging arrangement discussed – reaching a value of 166% at 10 psi. Intercoolers are much better adapted for use in marine engines where cool sea water is available to absorb the heat than for automotive-engine use.

THE AUTHOR: JOHN L. RYDE was assistant to the manager of engineering, when he left Fairbanks, Morse & Co. in 1940 to join the McCulloch Engineering Co. He is now chief engineer of McCulloch. Mr. Ryde received his B.S. degree in 1933, and his M.S. degree in 1934, from the University of Illinois.

THE past few years have marked a greatly accelerated development in increasing the specific output of the internal-combustion engine, and a widespread application of supercharging to many varied types of units has been undertaken. Paralleling the rapid advances in the aircraft field, the progress is particularly noticeable in the small-bore higher-speed four-cycle engines for powerplant, automotive, and marine applications where the demands of the armament program for high output in small packages have greatly stimulated the research. As may be expected, the greater part of this work is among the diesel and spark-ignition oil engine manufacturers in an effort to close the gap between the specific output of these units and that of the gasoline engine. The oil-burning engine has a well-justified attraction for military applications in the material reduction of fire hazard.

On this type of equipment, the positive-displacement supercharger, and almost exclusively, the Roots type, has become more and more popular due partly to its operating characteristics and partly due to its desirable features of production and application.

Recently a number of excellent papers and discussions have been presented comparing the merits of the various methods of supercharging and the problems incident to the supercharged engine. This material has been presented quite thoroughly and to go into detail on this subject would be needless repetition. The three most prevalent arrangements – that is, the exhaust-driven turboblower, the engine or mechanically driven centrifugal unit, and the engine-driven positive-displacement blower unit – have their own spheres of activity, and the first two have reached a high state of development in the aircraft and larger diesel-engine field.

Advantageous Characteristics

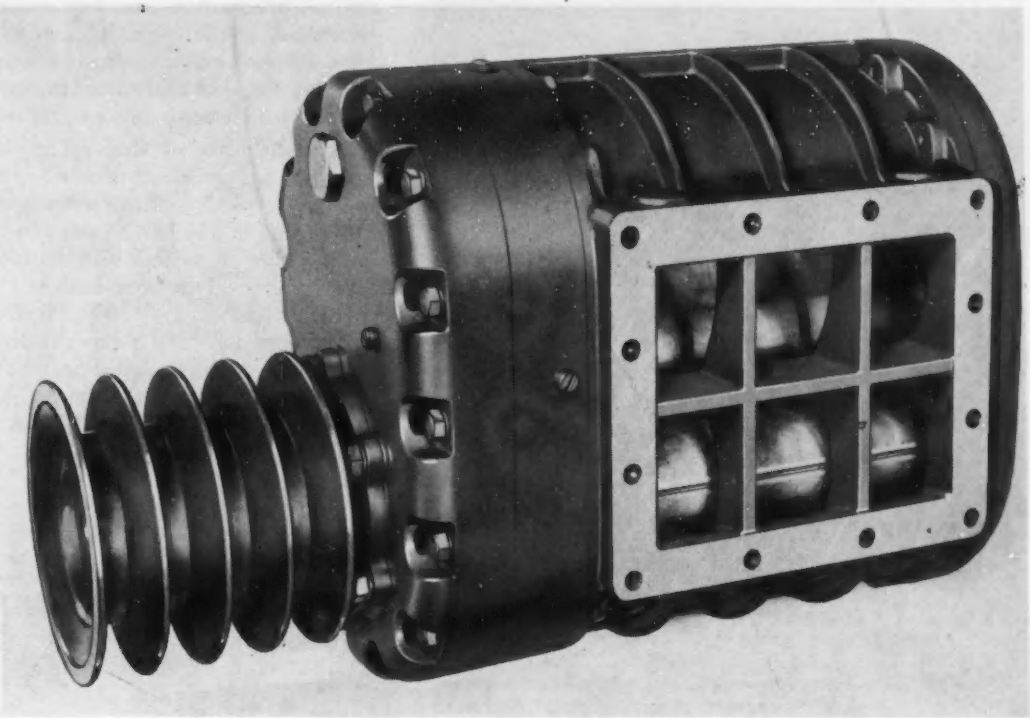
The positive-displacement supercharger finds its most advantageous application on automotive-type engines. The operation of these blowers is characterized by a fixed displacement of air per revolution, and delivery is affected only to a small extent by pressure variations. Thus, at reduced speeds, the supercharger continues to deliver extra air to the engine, and for such services requiring a high torque at lower speeds, this characteristic is advantageous. It imparts the desirable "lugging" ability to the engine. This characteristic is in contrast to the engine-driven centrifugal units where the pressure is proportional to the square of the rpm and, consequently, falls off rapidly as the speed is reduced.

[This paper was scheduled for presentation at the 1942 Semi-Annual Meeting of the Society, which was ruled out because of transportation priorities; it was presented at the ASME 15th National Oil & Gas Power Conference, the SAE Diesel-Engine Activity Cooperating, Peoria, Ill., June 19, 1942.]

These superchargers have been applied also to the smaller constant-speed power unit engine where the speed characteristic is not important, and, in addition to this application, they are being used to a considerable extent on small marine diesel and gasoline engines. Here one

T

SUPERCHARGER



■ Fig. 1 – Model 4090 Roots supercharger

would expect the centrifugal unit to have the advantage due to the fact that its characteristic approaches that of the propeller power requirements versus speed. However, the complication of the centrifugal blower gearing, as contrasted with the simplicity of the Roots blower, offsets this advantage in the eyes of many.

From the standpoint of the user of the supercharger, the engine manufacturer, the most important points are the performance of the unit on the engine from both the standpoints of operating characteristics and reliability, the cost and weight of the equipment as proportioned to the gain in power output, the ease of adaptability to the engine, and the problem of service in the field. The supercharger manufacturer inherits all of these problems plus that of production.

■ Performance

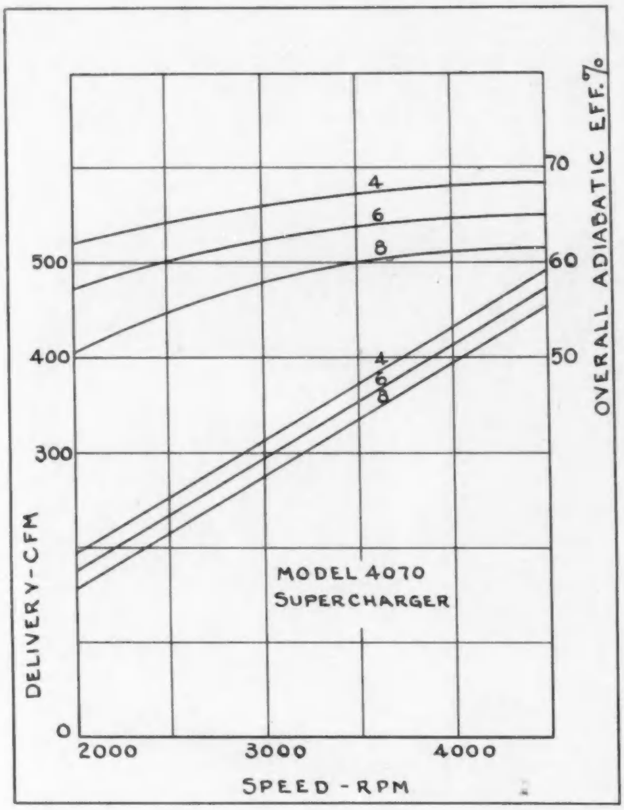
When considering any of the mechanically driven superchargers with regard to performance on the engine, the efficiency of the compressor becomes of primary importance. The power required to drive the unit must be deducted from the engine output, so that any saving here is reflected directly in the overall thermal efficiency of the combination. Likewise, the efficiency of the blower affects the amount of the supercharge or the percentage increase in brake mean effective pressure that can be obtained economically.

For instance, a reduction in supercharger efficiency on a given installation results in the delivery of less air for a constant blower horsepower input. The manifold pressure is lower, and moreover, the air delivered by the blower has a higher relative temperature, causing a further reduction in air density. The engine is only interested in the density of the air filling the cylinder as this is the factor determining the amount of fuel burned and the power developed.

Thus the combined effect of a lower manifold pressure and a higher air temperature causes a reduction in output of considerable magnitude. This effect is more apparent at or near the maximum output of the engine, but also appears in the thermal efficiency at more conservative ratings.

The various types of positive-displacement blowers may be classed into two groups – those which compress the air within the blower before delivery, represented by the vane or eccentric types, and those units which depend on the back flow of air in the discharge to compress the new charge delivered. The Roots blower is the best example of the latter class.

Of the two types of superchargers, the first type which compresses the air before delivery is the more promising from the standpoint of efficiency. The operation approaches that of the ideal air compression cycle, and the efficiency is theoretically higher. On the other hand, the Roots blower operates on a cycle having a "rectangular" indicator card where the air is compressed at constant volume before any delivery is made. Consequently, even a perfect Roots blower with no losses has an efficiency which is less than the adiabatic compressor, and the difference increases at higher pressures.



■ Fig. 2 – Efficiency versus speed characteristic – Roots supercharger

However, the vane type of compressor for supercharging applications has a very serious drawback in its mechanical complication and the lubrication problem. A number of modifications have been offered on the market, such as the Cozette, Centric, Zoller, and similar units, but all have the slidable or oscillating vane requiring lubrication and introducing additional friction. These wearing surfaces are open to the air enclosures, making it necessary to limit the flow of lubricant as much as possible, and the lubricant becomes contaminated easily. This is an undesirable hazard on the compression-ignition engines where sufficient excess oil passed by the blower may result in an engine runaway if the lubrication rate should suffer a maladjustment.

The Roots blower, shown in Fig. 1, in spite of its handicap in the theoretical efficiency of the operating cycle, has many redeeming features which fulfill the requirements of a supercharger. It is the simplest design, having only two moving parts. There is no lubrication required in the air chambers of the blower, and the only points at which friction occurs are the bearings, seals, and gears.

The unit is easily adapted to the engine, and the size is in keeping with its air delivery and speed characteristics. In most cases, a satisfactory installation can be made with a speed-up ratio between the engine and blower of from 1½ to 3:1 on the smaller engines and, as a result, the drive problem usually is not complicated.

The overall adiabatic efficiency curves of a small Roots blower are shown in Fig. 2. This unit incorporates some improvements in sealing, resulting in a higher volumetric efficiency. It will be noted that the adiabatic efficiency is almost on a par with the centrifugal blower where the efficiency ranges from 65 to 70%. This result is due to the high mechanical efficiency which compensates for the power loss of the cycle. Moreover, the curves show another characteristic of particular importance in that they are fairly flat, and the blower can be operated over a wide range of speeds with almost equal performance.

From the point of view of the supercharger manufacturer, this feature means that a fewer number of different sizes are required to cover a wide range of air deliveries, as the speed of any one blower can be varied to provide a sizable range. A further simplification can be made when a blower of a given cross-section is designed in different lengths of impellers. By selecting certain lengths between approximately 1½ and 3 times the center distance, a series of blowers can be designed to cover still a wider range. All of the series use the same end plates, gears, bearings, and seals, and the rotors and housings differ only in the one dimension of length.

Model	Center Distance, in.	Length, in.	Displacement at 60% 100% Rpm Rpm
4055	4.0	5.5	253- 422
4070	4.0	7.0	322- 537
4090	4.0	9.0	415- 690
4012	4.0	12.0	552- 920
5575	5.5	7.5	507- 844
5590	5.5	9.0	608-1013
5511	5.5	11.0	743-1239
5514	5.5	14.0	945-1576
5518	5.5	18.0	1215-2025
7512	7.5	12.0	1075-1795
7516	7.5	16.0	1435-2392
7521	7.5	21.0	1825-3145

The foregoing table shows three such series selected to cover air requirements from approximately 150 to 2500 cfm. Three center distances are used, 4, 5.5, and 7.5 in. The lengths were established in an approximate geometric progression, and each unit overlaps its neighbor in capacity, so that there are two blowers for almost any air requirement in the range. This feature is important as a supercharger for a heavy-duty engine in continuous service should operate at a lower speed than that for a high-output intermittent-duty unit.

All of this is accomplished in 12 sizes, but only three different sets of standard parts. This arrangement permits a substantial saving in production costs as well as a great simplification of the service problem. It will be seen that these same considerations apply to any of the positive-displacement designs where the length may be varied for different capacities.

One of the first questions on the application of the supercharger is how much additional power can be gained, and what kind of overall engine performance can be expected. This has been discussed at some length from a theoretical point of view[1, 2], but it should be noted that there are other considerations which must be taken into

[1] See *Diesel Power*, October, 1940, and January, 1941: "Supercharging," by Ralph Miller.

[2] See SAE Transactions, November, 1941, pp. 481-487: "Mechanical Supercharging of Diesel Engines," by H. L. Knudsen.

account. Speaking now of the diesel engine where the greatest amount of development work is being done, the performance depends a good deal on the design or type of engine and also on the service for which it is intended. For instance, in marine service we have cold sea water available and thus an opportunity to cool the air between the blower and the engine.

As will be shown later, this cooling makes a great difference in the final results. However, on an automotive engine for trucks, tractors, and other mobile equipment, we have no sink for the heat and, at the present time, these units must operate with no cooling other than normal radiation.

Coming to the design of the engine, the question of scavenging by valve overlap enters. Engines having an open-head construction or a combustion-chamber arrangement with a considerable volume in the cylinder, such as the Lanova, can make use of valve overlap to clear the cylinder of residual gases. This makes an appreciable difference in the economics of supercharging as compared with an engine whose combustion chamber is separated from the cylinder, and no valve clearance is available without seriously upsetting the basic design.

With these thoughts in mind, it is interesting to show the somewhat diversified variations now in practice and to compare the results that are possible with present-day supercharging equipment.

The sole function of the supercharger is to deliver air of increased density to the cylinder. In effect, it is the first stage of a two-stage air compressor, and the same considerations apply. Consequently, the operation is established by the fundamental gas laws, and can be expressed as follows by equating the conditions in the cylinder with those at the suction of the blower:

$$\frac{P_c V_c}{T_c} - \frac{P_{exh.} V_{cl.}}{T_{exh.}} = \frac{P_1 V_1}{T_1}$$

where P_c, T_c represent conditions in the cylinder. V_c is the total volume of the cylinder.

$P_{exh.}$, $T_{exh.}$ are the exhaust temperature and back pressure. $V_{cl.}$ is the clearance volume.

P_1, $V_1 T_1$ are the outside air conditions.

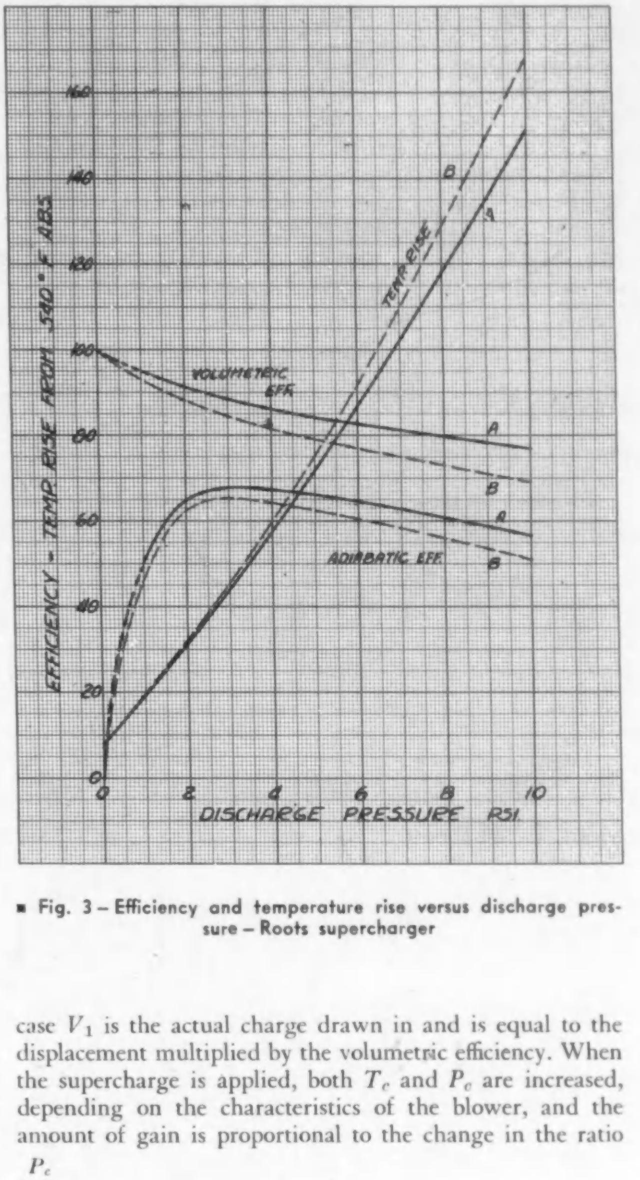
$$V_1 + \frac{V_{cl.}\, P_{exh.}\, T_1}{P_1\, T_{exh.}} = \frac{P_c V_c T_1}{T_c P_1}$$ and, for fixed external conditions,

$$V_1 + V_{exh.} = \text{Const.}\ \frac{P_c}{T_c} \qquad (1)$$

where $V_{exh.} = \frac{V_{cl.}\, P_{exh.}\, T_1}{P_1\, T_{exh}}$ the residual gases reduced to external conditions.

This relation demonstrates the equal importance of the charge temperature, T_c, as compared with the cylinder pressure P_c. Since the volume of fresh air entering the cylinder, V_1, is the measure of indicated power, the reduction of T_c assumes considerable importance, even more so than the increase in the pressure, P_c. Increasing P_c to increase V_1 requires more blower horsepower, and this power must be subtracted from the output in the case of the mechanically driven blower.

This relation applies to any normal unsupercharged engine with no effective overlap or scavenging. In this case V_1 is the actual charge drawn in and is equal to the displacement multiplied by the volumetric efficiency. When the supercharge is applied, both T_c and P_c are increased, depending on the characteristics of the blower, and the amount of gain is proportional to the change in the ratio $\frac{P_c}{T_c}$.



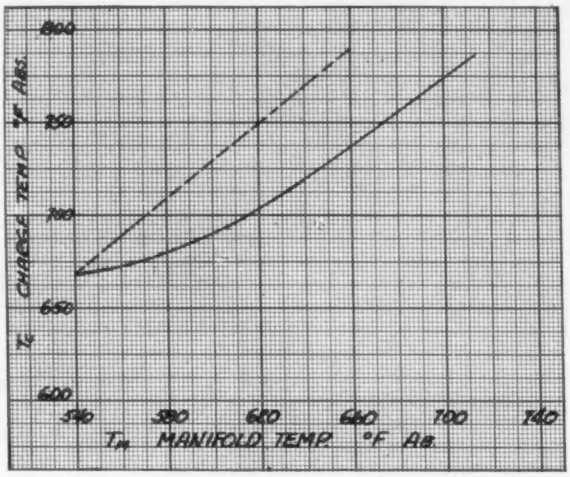
■ **Fig. 3 – Efficiency and temperature rise versus discharge pressure – Roots supercharger**

Fig. 3 shows the performance characteristics of a positive-displacement Roots blower as a function of the discharge pressure. From the temperature curve shown it is possible to determine the engine manifold temperature for any supercharging pressure. When the supercharger is applied, it delivers a fixed volume V_1 determined by the blower speed and pressure, and thus, knowing P_c and V_1, the temperature of the charge T_c can be calculated by the foregoing relations.

From this value, the temperature rise, t, due to heating from the cylinder walls and residual gas can be determined, for $t = T_c - T_m$, where T_m is the manifold temperature.

A number of observations have been made on supercharged engines with no scavenging, and the calculated values of the charge temperatures determined. Fig. 4 shows the values obtained on an engine with a piston speed of 1400 fpm, plotted on the basis of inlet manifold temperature. The curve is rather surprising in one respect in that the temperature rise is rather slow at first. It must be remembered, however, that T_c is not a function of T_m

alone. At low supercharging pressures and a constant air fuel ratio, the cylinder wall and valve temperatures will remain much the same, and the quantity of heat transferred approximately constant for the same temperature difference. However, we are admitting a larger amount of fresh air and thus the temperature rise, t, due to heat from the walls will be reduced. At higher pressures and temperatures, the increase in volume becomes proportionally less and the higher cycle temperatures apparently cause the curve to rise more rapidly and approach the slope of the dashed constant temperature rise line.



■ Fig. 4 – Charge temperature versus inlet manifold temperature – 1400 fpm piston speed

With this information it becomes possible to compute the supercharged performance at any pressure level using the blower with the efficiency curve A shown in Fig. 3. The values of V_1 for various supercharger pressures may be determined and the ratio $\dfrac{V_1}{VE \times \text{Displacement}}$ is the indicated power increase at a constant air-fuel ratio and indicated thermal efficiency, where VE is the volumetric efficiency of the unsupercharged engine.

But it is not a true picture of the actual conditions to assume a constant thermal efficiency at higher supercharging pressures. In practice the volumetric compression ratio must be reduced at higher pressures to avoid excessive compression and maximum pressures, and consequently, the indicated thermal efficiency is reduced. To obtain the true characteristics for a constant air-fuel ratio, the imep must be corrected for the change in thermal efficiency at lower compression ratios.

The following table shows the effect of reducing the compression ratio for various supercharging pressures in order to maintain an approximately constant peak compression pressure. The ratios are weighted slightly to provide a higher pressure at increased supercharge and are based on a polytropic exponent of 1.30. Too low a compression affects the engine starting seriously, as the supercharge is very low at cranking speeds, and special compensations are often required.

Manifold Pressure, psi	Compression Ratio	Clearance Volume cu in.	Thermal Efficiency Factor
0	16.5	6.45	1.000
2	15.3	7.00	0.972
4	14.2	7.58	0.953
6	13.3	8.13	0.932
8	12.5	8.70	0.913
10	11.9	9.20	0.897

The thermal efficiency factor is computed from the mixed-cycle formula assuming a constant cut-off and that the actual thermal efficiency will be reduced in proportion to the theoretical efficiency. While this may not be strictly true, the values will be quite close to these shown, since the air-fuel ratio is constant. The increase in imep is given then by the following:

$$\text{IMEP}_{\text{sup.}} = \text{IMEP}_{\text{std.}} \times \frac{T_1}{VE\,P_1}\left[\frac{V_c\,P_c}{T_c} - \frac{V_{\text{cl.}}\,P_{\text{exh.}}}{T_{\text{exh.}}}\right] \times \text{Thermal Efficiency Factor}$$

The overall net gain is:

$$\text{BMEP}_{\text{sup.}} = \text{IMEP}_{\text{sup.}} - \text{Friction MEP} + (0.8\,P_m - \text{Blower MEP}).$$

where P_m is the manifold pressure.

The blower MEP is computed from the efficiency curve of the supercharger and is a function of V_1 and P_m. It will be noted that a value of $0.8\,P_m$ is used as the effective pressure during the charging stroke of the engine cycle. From motoring tests with and without the supercharger, this result was obtained and, while it is subject to question due to the influence of other conditions in such a test as the changes in engine friction, the factor is conservative.

By this method, the effect of supercharging the engine at 2, 4, 6, 8 and 10 psi manifold pressure was calculated, assuming that the charge temperature followed the values shown in Fig. 4. The compression ratio and clearance were corrected for each value of the manifold pressure, P_m, and the imep corrected for the change in thermal efficiency. The characteristics of the unsupercharged engine are as follows:

Bmep	85 psi
Mechanical Efficiency	80%
Volumetric Efficiency	82%
Compression Ratio	16.5:1
P_1	14.7 psi
T_1	540 F abs. (80 F)
$P_{\text{exh.}}$	14.9 psi (0.2 psi back pressure
$T_{\text{exh.}}$	1260 F abs. (800 F)
P_c (end of suction stroke)	14.5 psi or ($P_m - 0.2$)
T_c (computed)	668 F abs. (208 F)

The foregoing conditions are representative of a power unit with a piston speed of 1300 to 1400 fpm, or an automotive unit at the point of maximum torque.

From the indicated thermal efficiency and the mechani-

al efficiency at each point, the variation in fuel consumption was determined and plotted as a percentage of the unsupercharged fuel rate. The engine friction was assumed to be constant.

Following the method of Miller[1] the cylinder volume V_c and clearance volume $V_{cl.}$, may be expressed as a percentage of the engine displacement simplifying the calculations. Then:

$$V_c = 106.45$$
$$V_{cl.} = 6.45$$

Fig. 5 shows the results of the calculations with the bmep expressed in per cent of the standard rating of 85 psi. The upper solid line A shows the increase based on a constant indicated thermal efficiency, while the heavier curve A shows the actual gain that may be accomplished with the correction for thermal efficiency.

It is interesting to note that the greatest gain per pound increase in manifold pressure is made at the lower pressures. This arrangement is a good example of the law of diminishing returns. Beyond 7 psi pressure, the gains are small, and the brake thermal efficiency as shown by the fuel-consumption curve is affected more seriously.

Curves B on Fig. 5 show the effect of blower efficiency. Referring back to Fig. 3, the dashed curves B represent a blower with an adiabatic efficiency approximately 5% lower at 10 psi discharge pressure. By repeating the same calculations with these values, the engine performance was again determined. It will be noted that the performance is affected quite markedly, and that no gain in horsepower output is made when the manifold pressure is increased from 8 to 10 psi. This effect at high pressures has been observed on a number of engine tests using older-type blowers with lower efficiencies. Increasing the supercharger rpm to obtain more power resulted in no additional horsepower and, if carried still further, the output of the engine actually fell off.

The increase in output shown by Fig. 5 represents very closely the actual performance that has been achieved. The usual "mark-up" for this supercharger arrangement is from 25 to 30% with manifold pressures up to 8 psi, which represents just about the economical limit of supercharge, and a high blower efficiency is required.

■ Scavenging

On engines having sufficient or available clearance for valve overlap in the combustion chamber, a substantial gain in output can be made by clearing the cylinders of the residual gases, thus reducing T_c. In addition, the volume previously occupied by the residual gas is added to V_1, increasing the amount of the fresh charge. This result is accomplished by blowing an extra amount of air through the clearance volume during the overlap period.

The calculation of the engine performance can be made by assuming that the valve overlap is adjusted until a certain volume is blown through the engine at a given manifold pressure before the exhaust valve closes and traps the charge. Obviously, for very low pressures and high speeds, this overlap may be impossibly long but, for the purpose of comparison, the foregoing assumption can be made.

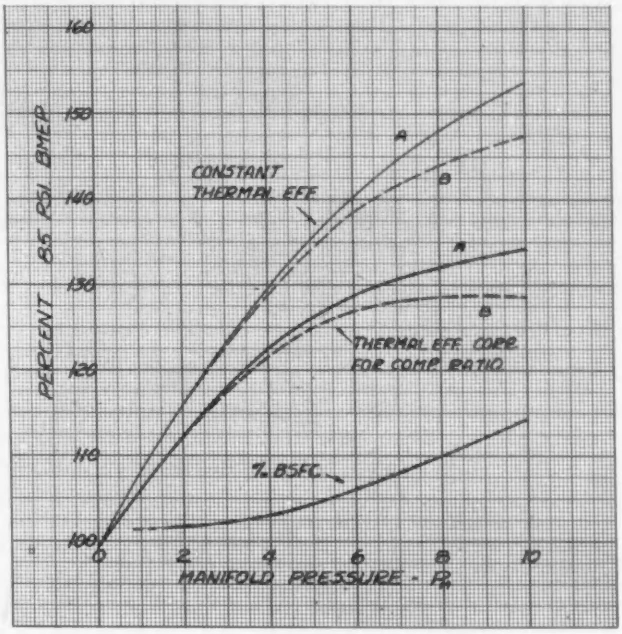
In the formula for the imep the term for the residual gases drops out. Also the extra volume for scavenging must be added to V_1 in determining the blower mep.

The expression for the imep becomes

$$\text{IMEP}_{\text{sup.}} = \text{IMEP}_{\text{std.}} \left(\frac{T_1}{VE \cdot P_1}\right)\left(\frac{V_c\ P_c}{T_c}\right)$$

$$V_1 = \left(\frac{T_1}{P_1}\right)\left(\frac{V_c P_c}{T_c}\right) + \text{Volume of Scavenge Air}$$

For the purpose of comparison, we will use an additional scavenge volume of twice the clearance volume of the engine. The charge temperature, T_c, is computed from the normal temperature rise in the non-scavenged engine by correcting for the heat of the residual exhaust



■ Fig. 5 – Per cent power increase in unscavenged engine

gas and the increased volume of air. If we assume that the scavenge air admitted first completely clears the residual gases from the cylinder head when it is expelled, the cylinder is filled with fresh charge. The total amount flowing through and into the cylinder also cools the walls and intake valve, causing a further reduction of the charge temperature.

Any method of calculating the cooling effect in the scavenged cylinder is, of course, subject to question as it is difficult to evaluate just what happens. From published data on air-cooling and heat-transfer research, the heat-transfer coefficient from a hot surface to air is proportional to the mass velocity of the air flowing past the surface raised to a fractional power, and the exponent varies from 0.4 to 0.8 depending upon the conditions. Thus, for fixed incoming air and cylinder-wall temperatures, the average temperature rise of the air is reduced as volume of air is increased.

However, it is reasonable to assume that, in the engine, the initial scavenge air will take up a greater amount of the heat as it contacts and chills the high temperature surfaces first, and that the succeeding mass of air has a corresponding lower temperature rise.

With these factors in mind, the cooling effect was estimated by assuming that the heat transfer was equal to that occurring in the non-scavenged engine at the same conditions, and that temperature rise in entering the cyl-

inder is reduced in proportion to the increased mass of air. Thus, if the temperature rise in this case is t_2, its value is given by

$$t_2 = \frac{V_1'' + \text{Scavenging Volume}}{V_1}\left[t - \frac{V_{\text{exh.}}}{V_1}(T_{\text{exh.}} - T_c)\right]$$

where V_1'' is the volume trapped in the scavenged cylinder
and V_1 is the volume trapped in the unscavenged engine at an equivalent pressure
t is the temperature rise of charge, $(T_c - T_m)$, in the unscavenged engine.

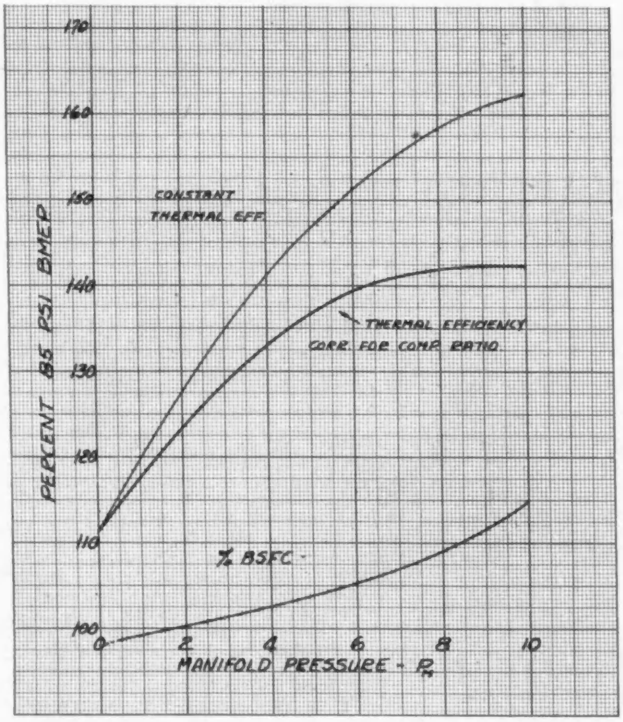
Likewise, all the terms in the bracket are those applying to the unscavenged engine at an equivalent supercharge pressure.

Thus $T_c = T_m' + t_2$ for the scavenged engine.

It will be noted that t_2 is established by V_1'' which is, in turn, a function of t_2. However, the value of t_2 can be readily established by approximations. By setting t_2 equal to the bracketed term just given, V_1 is determined by Equation (1). Using this value of V_1, a new value of t_2 is determined, and so on. The second approximation is accurate within a few degrees.

Of the two factors affecting the charge temperature difference in the scavenged engine, the removal of the exhaust gases causes the greater reduction amounting to 22 F at an exhaust temperature of 900 F. The cooling effect of the scavenge air is of a lesser magnitude, 12 F, at a manifold pressure of 4 psi. Thus, the estimate of the cooling effect is conservative and, even if in error due to the method of computation, the effect on the final result is small.

In opposition to this factor, the assumption that the residual gases are removed completely is only theoretically possible. Although the residual gases have been displaced by three volumes of fresh air, two in scavenging and one in filling, there will still be some contamination, and the amount depends a great deal on the shape and porting of the chamber.



■ Fig. 6 – Per cent power increase in scavenged engine – excess air equal to twice clearance

Fig. 6 shows the result of scavenging the engine cylinders on the power output. The gain in bmep is increased by approximately 10% over that of the nonscavenged engine at the lower pressures. However, it is interesting to note that the difference is less at high pressures, and the curve has a pronounced flat top above eight psi. The extra air for scavenging costs dearly in blower horsepower at high pressures.

At lower manifold pressures, the operation is much improved over the unscavenged engine. Owing to the greater fresh charge for a given pressure, the mechanical efficiency is improved, and the fuel consumption reduced.

At 4 psi pressure the engine shows an increase of 33% in power and the fuel consumption is increased only 2.4%. This is about the most economical point, and it is borne out by the fact that there are two recent engine developments designed to operate at this pressure. The performance of these units is very satisfactory with power increases ranging from 26% to 30%.

■ Intercooling

When the fundamental equations are examined, it is obvious that a great saving can be made by reducing T_m, the manifold temperature, and consequently, the charge temperature. The usual method is the insertion of an air cooler between the supercharger and the engine. As mentioned previously, this is accomplished quite easily on the marine engine where cool sea water is available to absorb the heat. It is not so easily attained on other applications such as automotive units where there is no sink at a reasonably low temperature.

As may be expected, the output of the intercooled engine as a function of the supercharging pressure is dependent on the amount of cooling. Fig. 7 shows the characteristic of an engine fitted with an intercooler capable of maintaining the manifold temperature at 125 F. This, of course, requires a progressively larger cooler for higher supercharging pressures.

The combined effect of the intercooler and the use of scavenging permits the highest increase of any arrangement, reaching a value of 166% at 10 psi. Additional gains can be obtained at even higher supercharging pressures, and the changing slope of the characteristic is due principally to the change in thermal efficiency.

However, it should be noted that performance at low pressures is very similar to the scavenged engine as the effect of the intercooler is only evident at higher pressures. Below 5 psi pressure the intercooler is of little value and is prescribed only when the maximum output is desired.

The demand for high-output marine engines by the Navy has brought forth some new developments utilizing a high manifold pressure, scavenging by valve overlap, and an intercooler. Bmep's of 150 psi have been obtained with blower pressures of 11 psi, while other designs utilizing just the intercooler have given very satisfactory performance at 130 to 135 psi bmep.

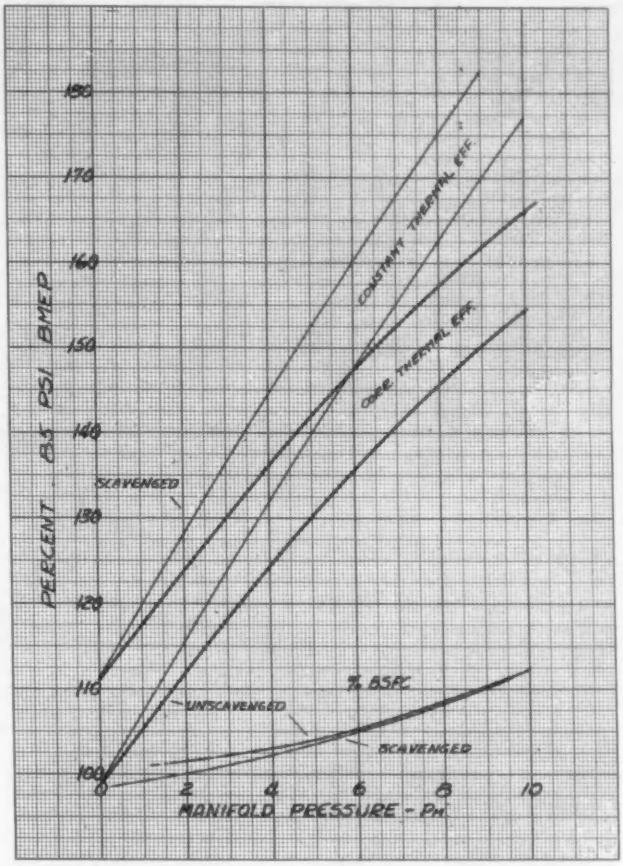
■ Need for Information

The preceding material is offered as a preliminary means of estimating the performance of a positive-dis-

placement supercharged engine and also as a yardstick of the practical limitations for the performance of the units with present-day equipment. As in any rational analysis, certain assumptions must be made but, in every case, values from actual practice were used in so far as possible.

It must be remembered that the characteristics will vary somewhat from engine to engine and will be affected by the size of the engine, piston speed, and other special considerations. For instance, an engine with a precombustion chamber having a narrow opening will have the charge temperature characteristics affected in that the residual gas will not mix readily with the fresh charge and will dissipate a greater amount of heat to the walls, contributing less to the charge temperature increase.

Certainly, additional test data on these points will be very welcome and will contribute a good deal to our present limited knowledge. Most of the present supercharger applications are made with as careful analysis as possible supported by past experience, but a good many unknowns still remain. From our position as supercharger manufacturer, we have an excellent opportunity to collect the general experience, but we are somewhat remote from the actual detailed research. Consequently, it will be of great value to hear more on some of the foregoing considerations from the engine manufacturers who can probably make far more substantial contributions.



■ Fig. 7 – Power increase with intercooler – manifold temperature maintained at 125 F

■ Blower Design

This discussion has emphasized the importance of blower efficiency, and it is not surprising that most of the development work is along these lines. Some new designs have been introduced recently such as the Lysholm and Whitfield blowers. These units are similar to the Roots unit in construction but, by using special helical rotors and triangular ports located at the ends of the housing, the air is compressed between the rotors before delivery, with a consequent increase in efficiency. The performance of these units is quite promising, but the manufacturing problems are multiplied manyfold, due to the specially shaped rotors.

Moreover, all of these units together with the Roots type share the same handicap. One of the most awkward problems to handle is that of clearance. The two rotors must not touch each other or the casing at any point while running, or galling will occur, ultimately damaging the supercharger. At the same time, the clearances around and between the rotors permit a leakage of air back from the delivery similar to a leaky piston in an air compressor. This leakage seriously affects the efficiency and must be held to a minimum in any type of machine.

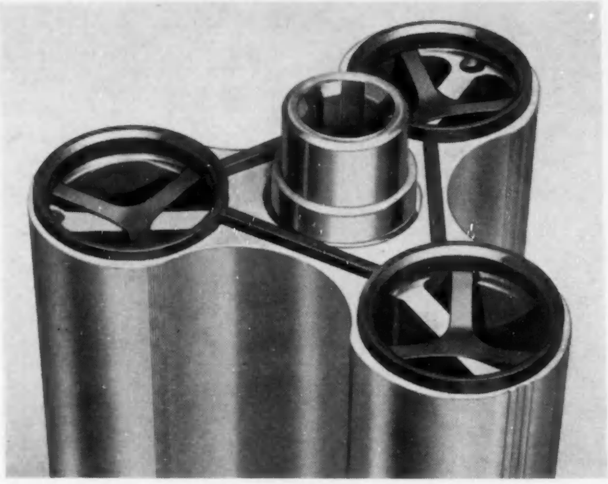
To determine the correct clearance for a universally adaptable supercharger is not an easy thing. The designer has to follow a very narrow path with unsatisfactory operation due to low efficiency on one side and disaster on the other. Then, too, the conditions vary from engine to engine, and a satisfactory clearance for one engine might be dangerous for another.

The worst offender in leakage is the axial clearance of the rotors and the end plates, and particularly in the area between the shafts. Here the movement of the rotors assists the leakage and practically paddles air back to the inlet. Unfortunately, seizure usually occurs at the ends due to the distortion of the outside case under load where the discharge side is hotter than the intake side. Also, on sudden load changes, the rotor expands before the housing, causing a transient reduction in clearance, and this factor must be accounted for in setting the distance.

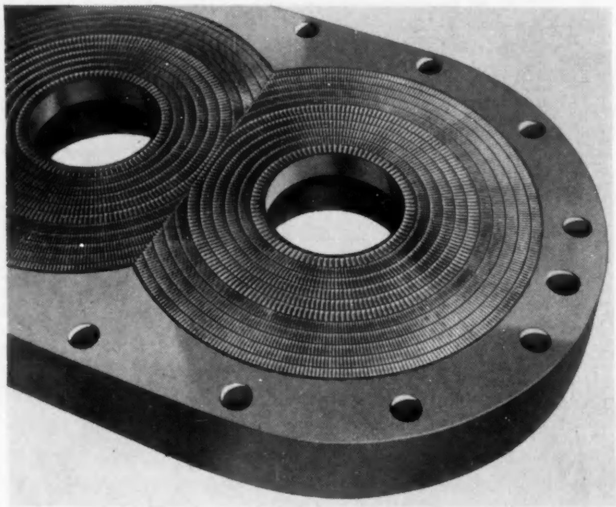
To eliminate these difficulties, various methods of reducing this leakage have been worked out, with some excellent results. In one method shown in Fig. 8, a series of carbon seals running in contact with the ground end plates was used to close the end clearance to zero. These seals are held in contact by light springs as shown and act in the same manner as the rings on a piston except that no lubrication is required. The material is a hard carbon compound containing graphite and lead and is similar to that used in electric motor brushes.

For best operation, the contour of the seal should be identical with that of the rotor, but this is difficult to attain and the compromise shown is simple and effective. This arrangement permits the use of safe end clearance and, at the same time, a sizable gain in overall efficiency is obtained. In addition to this advantage, the closing of this clearance solved another production problem – that of the variation of efficiency from unit to unit.

In spite of the very close tolerances on the rotor and housing lengths as well as the bearing positioning shoulders, the variation in end clearance is equal to the sum of these and represents a sizable proportion of the total end clearance. This variation affected the overall efficiency to such an extent that it was noticeable in the engine performance particularly at high bmep. The end seals elimi-



■ Fig. 8 – Carbon seals on supercharger rotor



■ Fig. 9 – Synthetic rubber seal bonded on end casting

nated this difficulty, and the variation from blower to blower in production was reduced to a small value.

Another method which has proved quite successful is the covering of the end casting faces with a thin coating of stiff synthetic rubber, molded, cured, and bonded onto the casting. The face is molded with a grid as shown in Fig. 9 and, in assembly, the rotor may actually touch the tips of the ridges. When the unit is run in on the test stand, the rotors wear the rubber grids down to the correct clearance without damage. The amount of wear is only a few thousandths of an inch and occurs very quickly. Later, if the blower is subjected to an unusual condition, closing the end clearances, the rubber coating takes the wear without damage to the machine.

■ Noise

Nearly all of the positive-displacement units are characterized by noisy operation, and the Roots blower is one of the worst offenders. This is the main reason for the helical rotor on certain designs which opens the bucket gradually to the discharge pressure. Likewise, the helix is such as to deliver the air smoothly without pulsations.

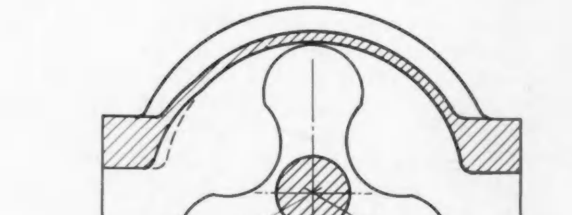
The difficulties of manufacturing the helix are obvious and, for this reason, an approximately equivalent modification of a straight rotor with a helical port was tested and adopted on the first units.

This arrangement gave some improvement and was similar in operation to the helical rotor. However, both were still very noisy, particularly at high discharge pressures.

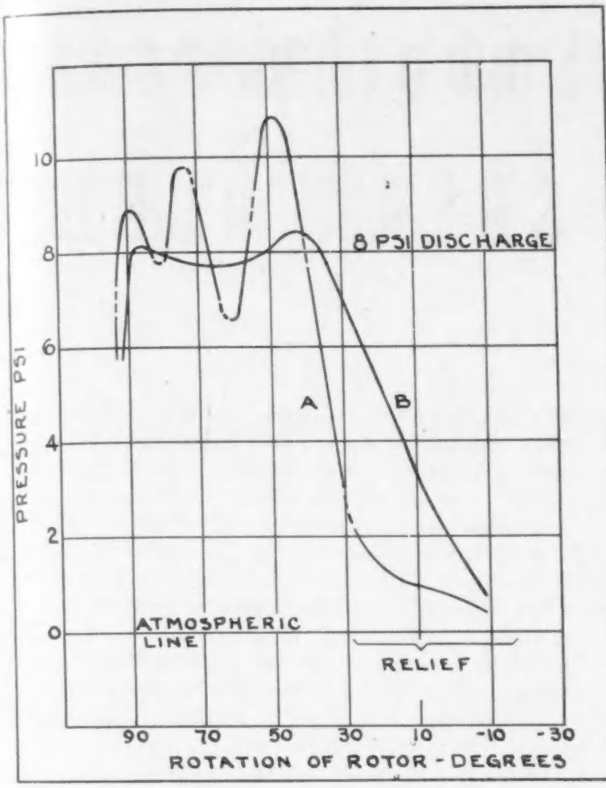
To obtain information on this action, a high-speed indicator was constructed to obtain pressure-time diagrams in the rotor bucket, and some very interesting facts were discovered. Even with the helical port, the back compression occurred much more quickly than was estimated. The rotor traveled a very few degrees before the back compression was completed with a bang. At high pressures only a small opening is required to permit sufficient air flow for completion of the process.

Much of the noise is generated by this shock and, if the back flow could be distributed over a greater time interval, it was believed that the effect could be reduced. In an attempt to accomplish this purpose, the housing was bored eccentrically on the inside diameter, permitting an increase in the clearance at the outside diameter of the rotor lobe as it approached the discharge port as shown in Fig. 10. The relief was increased in small steps and the effect observed on indicator diagrams until the pressure curve showed the minimum rate of pressure rise.

The indicator diagram, Fig. 11, shows the difference between the straight port and the relieved housing. Curve *A* has an abrupt pressure rise and it should be noted that high velocity created a surge in a bucket at a high frequency. The amplitude is large, and this sound could be heard distinctly as a high-pitched howl above the normal blower frequency of six pulses per revolution. The dia-



■ Fig. 10 – Diagram showing relief cushioning shock of back compression



■ Fig. 11 – Indicator diagram of pressures in rotor enclosures

gram shown covers the travel of one lobe and the action is repeated six times per revolution.

Curve *B* shows the result of the relief in the housing. The rate of pressure rise has been reduced, and the oscillation of the back flow materially lessened. The blower operated more quietly and could be muffled easily to a satisfactory level with a standard intake silencer. To show how quickly the back flow occurs, it is interesting to note that the amount of increased clearance or relief for best operation is very small, slightly over 1/16 in.

In connection with the characteristic sound of the blower, silencers of the "straight-through" type are very effective in reducing the noise level due to the high frequency. In fact, lining the walls of the inlet piping with suitable sound-absorbing material is effective in cutting down the noise. The high frequency also permits the use of resonant-type silencers with comparatively small bulk.

■ Conclusion

The foregoing material serves again to emphasize the need for a better air compressor which will meet the size, weight, and cost requirements of a supercharger. A number of people are working on the problem, and we may hope to see some considerable advances in the future due to the stimulus provided by the national emergency.

In closing, one might add that here is a golden opportunity for someone's inventive ability. The design of a light-weight, high-speed, air compressor with a high efficiency that can be manufactured at a reasonable price will find a welcome home. However, it seems to be something like the problem of the infinitely variable transmission and is not as easy as it appears.

Invention for Victory

SINCE human beings engaged in mortal combat, those with the best weapons and smartest ideas usually emerged victorious. This has been true for thousands of years. In modern warfare, it is still true.

The invention of gunpowder made all previous weapons impotent. Smokeless powder and rifled guns greatly enhanced the effectiveness of those who possessed them. Ericsson's invention of the armored revolving turret made all previous types of naval vessels obsolete. The submarine has certainly changed orthodox naval tactics.

In more recent times, the development of the airplane into a combat vehicle and the development of the caterpillar tread into such a terrifying monster as the modern tank have completely changed our mode of warfare.

In war time, the number of inventions and suggestions submitted by the public for Governmental use increases to such an extent that some special agency is clearly needed – an agency whose *only* job is to give immediate and sympathetic attention to every invention, suggestion or idea which might have any possible value in our war plans. The National Inventors Council was created for that purpose. It has been in active operation since about October 1, 1940. Results to date show that it *is* of very real value to the military services and to others.

Its value, however, is in almost direct proportion to the number of contributions which come from the trained minds of those who comprise the membership of our technical professions – particularly the members of this Society.

In January, 1942, the National Inventors Council had just finished a little more than a year of active operation. During that time, more than 40,000 cases had been examined by the Council staff. It is important that this flow of worth-while inventive ideas be not only continued, but increased if possible. For this reason, I bespeak your interest in what our Council is organized to do and make full use of its facilities.

The Council receives inventions, suggestions, and just plain ideas at its Washington, D. C., headquarters, by mail and through the personal calls of inventors. Many inventors seem to think their ideas are so important that no one but the President should know about them. Others send their inventions to Senators, Congressmen, Members of the Cabinet and to ranking officers of the Army or Navy, no doubt hoping for preferential treatment. But all are forwarded to our office for primary examination. Each invention, suggestion, or idea received is given careful examination and evaluation. Those appearing meritorious are then referred to the appropriate Technical Committee Chairman for his consideration and report. An invention, thought to be of value, is forwarded to the Army or Navy, or other appropriate defense agencies, for their consideration – and ultimate adoption if they are found acceptable. In such event, the appropriate department of the Army or Navy then deals directly with the inventor in making the necessary arrangements for the use of his invention. The Council itself does not consider the question of compensation or of contracts between inventors and the Army or Navy.

Excerpts from the paper of the same title by Col. L. B. Lent, chief engineer, National Inventors Council, presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 13, 1942.

Engine Smoothness and SHEAR RUBBER

by PAUL C. ROCHE
Field Engineer, Lord Mfg. Co.

IN that vibration commonly exists in rotating or reciprocating machinery, vibration engineers work either toward eliminating it at the source or to minimize its transfer to adjacent structures. This paper is intended to deal with the subject in connection with internal-combustion engines in general, and with a particular accent on automotive-type installations. It is quite true that much can be done in the way of eliminating vibration at the source although in commercial manufacture the logical extent of doing so becomes a compromise with the cost of carrying the correction to approach the ideal. Moreover, to be consistent with this degree of modernism and production thoroughness, the designer should pare powerplant weight to the bone, increase compression ratio, and incorporate efficient spark control, all of which will, in turn, augment the vibration problem. Granted then that we will continue to experience problems of minimizing vibration transfer it is to be shown: (a) that the presence of a resilient nonmetallic medium between powerplant and supporting structure is beneficial; (b) that rubber is well suited to this service; and (c) that shear-loaded metal-bonded rubber is its most effective manner of application.

[This paper was scheduled for presentation at the 1942 Semi-Annual Meeting of the Society, which was ruled out because of transportation priorities.]

To come face to face with the problem at the outset of this discussion a reasonable question is: "What are the causes and frequencies of the predominant vibrations in automotive-type engines?" or, in other words: "What conditions are to be combated by rubber mountings?" The powerplant, as a resiliently mounted mass, has six degrees of freedom, three of which are of a linear (translational) character, in separate planes, and the other three are of a rotary (torsional) character about three separate axes. The most easily pictured are the vertical translational case in a plane running the length of the engine and the torsional about an axis running the length of the engine.

A resiliently mounted powerplant, besides having various degrees of freedom, may, in each one of these various "modes," exhibit simultaneously and continuously several frequencies of vibration (disturbing frequencies) for any constant engine speed. For instance, any unbalanced condition of the rotating parts of the crankshaft system excites a translational disturbance of first order (one times engine speed), often referred to as a "primary" disturbance. Also the "inherent unbalance of the 4-cyl engine" excites a

THE purpose of this paper is to acquaint the engineer with the factors involved in the adaptation of resilient mountings to automotive powerplants. The benefits to be gained by the proper use of shear-loaded bonded rubber mountings are shown to be vibration and noise isolation, shock protection, and provision to accommodate structural twist without straining engine parts. Serving to disprove a number of erroneous impressions, this treatment points out, for instance, that it is not necessary to tolerate instability of the powerplant in order to realize a high degree of vibration absorption.

Representing considerable on-the-job experience in this type of work, particularly with commercial-type engines, the material in this paper forms a basis on which an engineer should be able to analyze and lay out his own installation. The author emphasizes the necessity for detailed consideration of such factors as disturbing frequencies of vibration, engine configuration, structural characteristics, and conditions of operation.

★ ★ ★

THE AUTHOR: PAUL C. ROCHE (J '39) is field engineer of Lord Mfg. Co., specialists on the design and manufacture of bonded rubber shear-type mountings and other such functional rubber parts. An automotive engineering graduate of Yale University in 1936, Mr. Roche spent four years as a sales engineer in the automotive mechanical rubber division, Firestone Tire & Rubber Co., prior to joining Lord Mfg. Co. in 1940. His present work is concerned principally with Army trucks, tanks, and other military equipment.

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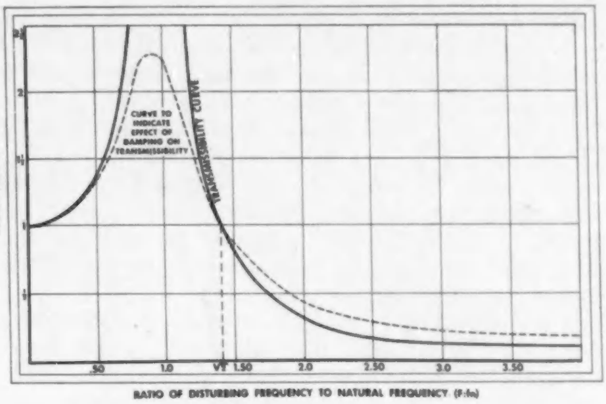
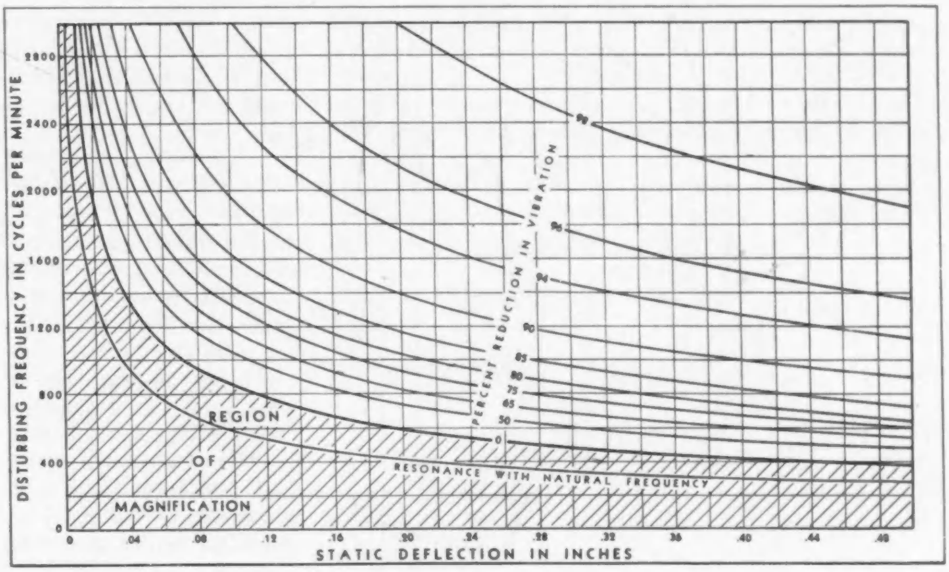
Protection through MOUNTINGS



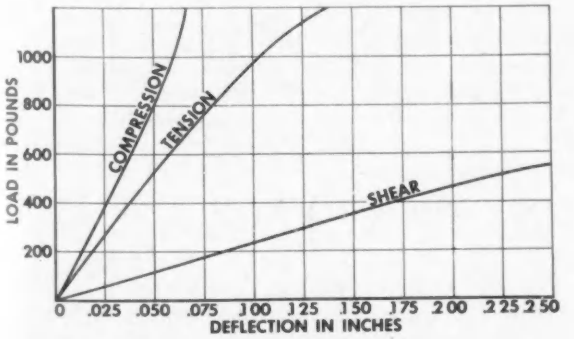
■ Fig. 1 – Relation between transmissibility factor and frequency ratio

translational disturbance of second order (two times engine speed), known as "secondary" disturbance, the cause of which can best be described as a shifting of the center of gravity as the engine rotates. In a 4-cyl engine this is the most severe vibratory condition. These translational disturbing forces are known as inertia effects and build up as the square of the speed.

A torsional vibration can be pictured as a rotational trembling about an axis along the length of the engine. Such a tremble may be caused by the individual combustion (power) impulses of the engine intermittently tending to rock the engine in a direction opposite to crankshaft rotation, and this happens to be the most severe torsional vibration; and, excepting the second-order translational disturbances in the normal 4-cyl engine, this is in most engines the most pronounced vibrating condition of any mode. The frequency of this torsional for the common 4-cycle engine is engine speed times half the number of cylinders, corresponding to second order for a 4-cyl engine, third order for a 6-cyl engine, and so on. These power impulses have a component occurring at engine crankshaft speed, which accounts for a first-order torsional vibration effect. Moreover, a misfiring cylinder in any 4-cycle engine sets up a torsional vibration occurring once every other crankshaft rotation, namely half order.



■ Fig. 2 – Mounting efficiency in terms of disturbing frequency and static deflection



■ Fig. 3 – Load-deflection curves for the same bonded sandwich specimen under compression, tension, and shear loading

If an engine were operating out in space, it is readily

recognized that the vibration transferred to the ground would be nil. Naturally, it is impossible to realize this ideal condition with an engine mounted on a framework but, by connecting through soft mountings, we aim to take a substantial step in that direction. Naturally a resilient mounting will sustain a steady deflection due to the weight of the engine supported; then, as the engine sets up vibratory impulses, an oscillating motion occurs on both sides of this statically deflected position. The effect of a vibratory force in the engine is to accelerate the entire powerplant in the instantaneous direction of the force, and it is resisted but little by the comparatively soft resilient mountings. But, before the distance through which the powerplant has moved becomes very great, the vibratory force has changed its direction; accordingly, the only part of the disturbing force transmitted to the frame is that small amount necessary to have deflected the resilient mountings a distance equal to the engine's motion. By an intelligent choice of static deflection rating in line with the disturbing frequencies encountered, the possible degree of vibration isolation is tremendous.

Every resilient body has its own natural period of vibration, which is the speed at which it will oscillate before coming to rest when subjected to a single impulse. The fundamental expression for this natural frequency in cycles per sec is

$$F_n = \frac{1}{2\pi}\sqrt{\frac{K}{M}}, \text{ where } M = \text{mass in slugs}$$

attached to mounting (whether supported vertically or otherwise) and K = spring rate of mounting (force required to deflect it unit distance, lb per ft) observed at the statically deflected position. In engine-mounting problems the engine weight is normally supported vertically by the mountings, and this weight is constant for a given installation, so the formula is simplified as follows

$$F_n = \frac{1}{2\pi}\sqrt{\frac{32.2}{D}}, \text{ where } D = \text{deflection in ft}$$

resulting from supporting the mass M on a mounting of translational spring rate K. Accordingly, static deflection offers a convenient means of computing natural frequency, provided such deflection is caused purely by weight supported.

This formula can be used to compute horizontal as well as vertical natural frequency by substituting in it an "apparent" deflection which can be visualized as the amount the rubber would deflect if actually supporting the same weight on a unit of stiffness corresponding to the spring rate in the horizontal direction.

The relationship between this natural frequency of vibration and the disturbing frequency of vibration from the engine is expressed:

$$T = \frac{1}{\left(\dfrac{\text{Disturbing frequency}}{\text{Natural frequency}}\right)^2 - 1},$$

where T is known as *transmissibility* factor or that portion of the engine's vibratory force that is conducted through the resilient mounting to the framework. In other words, when T has a decimal value only a part of the vibratory force reaches the framework, and when T = Unity, all the vibratory force is conducted through the mounting. Only for "frequency ratios" greater than $\sqrt{2}$ does a reduction of vibration occur. It should be noted that, when the natural frequency equals the disturbing frequency T = infinity, these cases are known as "resonance," often referred to as "critical" speeds of an installation, and should be avoided by proper design.

The curve showing the relation between transmissibility factor and frequency ratio is shown on Fig. 1. During resonance the natural frequency of the mounted system responds in step with the disturbing frequency, theoretically causing the amplitude of vibratory motion to build up to an infinite value. Particular care should be taken to insure that resonance occurs outside the operating range of speeds for the engine. If this is not practical, it should at least be made to occur at some speed known to be definitely transient, not a speed at which the engine may operate continuously, such as idling speed, cruising speed, top speed, or "stall speed" in the case of a fluid-drive installation.

■ Damping Characteristic

When this critical speed occurs, the only factor that actually keeps the amplitude from becoming infinite is the internal friction characteristic of the resilient mounting; this quality is technically known as hysteresis, and is said to have a "damping" effect on the system. The damping effect is markedly present in sluggish materials like cork or felt, is practically absent from steel springs, and exists in rubber to a limited extent. Though this characteristic is beneficial at the not-commonly-encountered resonant speed, it detracts from performance in the normal operating range (at high ratios of disturbing frequency to natural frequency) and, for this reason, rubber has been chosen as a vibration-isolation mounting in modern equipment because it possesses only a desirable degree of damping which, in fact, is controllable in the compounding of rubber. The broken line of Fig. 1 shows the effect of damping on transmissibility; please note this effect takes on an entirely new complexion after the frequency ratio exceeds $\sqrt{2}$. Fig. 2 gives a group of interesting solutions of the transmissibility formula, which are especially convenient to use in that percentage of vibration isolation is expressed in terms of: (a) frequency of disturbing vibrations; and (b) static deflection of mountings under engine weight. Rubber is particularly effective also in the isolation of noise transfer through a structure because sound is in essence a high-frequency vibration; a steel spring of equivalent spring rate is not as effective toward sound reduction in that it becomes apparently solid when subjected to certain high-frequency impulses. Moreover, the speed of sound through rubber is but a small fraction of its speed of travel through steel, which fact further contributes to its noise-interrupting qualities.

These same facts apply to torsional vibrations, and the torsional natural frequency of vibration of an installation in cycles per sec is equal to $\frac{1}{2\pi}\sqrt{\frac{K_t}{I}}$, where I is the moment of inertia of the powerplant (in slug-ft squared) and K_t is the torsional spring rate; this K_t is figured by multiplying the translational spring rate times *distance from elastic center (in feet) squared* for each individual mounting and summing the products, giving the torsional spring rate for the entire installation. The elastic center is the point about which a couple would cause rotary move-

ment of the powerplant, sometimes known as the center of elastic resistance. Having so arrived at this natural frequency, the *transmissibility* formula just given applies fully and, in this case, torsional disturbing frequencies are substituted. It is recognized that accurate figures on moment of inertia are not always available, but such data can commonly be arrived at by observing torsional resonance in a trial installation.

Effect of Spring Rate

It will stand out from the fundamental formulas just given that spring rates of mountings are the criteria of mounting performance. Only in the case of shear-loaded rubber are these spring rates substantially constant throughout the deflection range which indicates a decided point in their favor. In compression- or tensile-loaded mountings the constantly changing spring rate causes the stiffness on the "additive" side of the statically deflection position to differ from that on the "subtractive" side. Accordingly a "hopping" tendency is experienced with these types of mountings due to the pronounced torsional vibrations which act downward to mountings on one side of the engine while acting upward to those on the opposite side. Moreover, for any bonded rubber specimen the deflection for a given load is far greater in shear than in tension or compression. It is also true that rubber loaded in shear shows a lower damping factor. For these reasons, it is possible to gain most effective vibration isolation by using rubber mountings in shear. These facts are borne out by Fig. 3 showing load-deflection curves for the same bonded sandwich specimen under compression, tensile, and shear loading.

We often hear it said that we have to tolerate powerplant movement if we are to isolate vibration. This statement has its limitations and well deserves discussion here. As just expressed, vibratory forces cause a continual fluctuation of mounting deflection on both sides of the statically loaded position. The resilient mounting, for maximum effectiveness, should possess freedom to permit such oscillation to the fullest extent that can be brought about by the most severe vibratory forces excited, and its spring rate should be constant throughout that range. To that extent movement is to be tolerated, but it is not necessary beyond that degree, and good design practice is to incorporate in the mounting a "snubbing" feature to restrict movement outside this range, which additional movement would necessarily be of a shock nature caused by driving over bumps, and so on. The vehicle chassis springs, not the engine mountings, are designed to cushion this character of movement. Fig. 4 illustrates the load-deflection characteristic for a shear mounting with provision for snubbing against shocks. The working range of this mounting is indicated; its effective use depends on accurate knowledge of static load supported so as to insure operating entirely within the "working" range as far as vibration isolation is concerned. It is not uncommon to clamp a snubbing device into place after the mounting has been deflected under static load. The fact that the rubber cannot then return to its originally non-loaded position does not interfere with the application of the deflection formula for natural frequency, provided that such snubbing interferes in no way with the free working range of the rubber for vibration isolation.

The term "ideal mounting" creeps into discussions of powerplant suspension and its explanation is not difficult; it simply means that the points of mounting and the "axis of oscillation" of the powerplant should lie in the same plane. This axis is the one about which the moment of inertia is a minimum. A reasonably accurate method of establishing this axis in practice is to locate it as the line

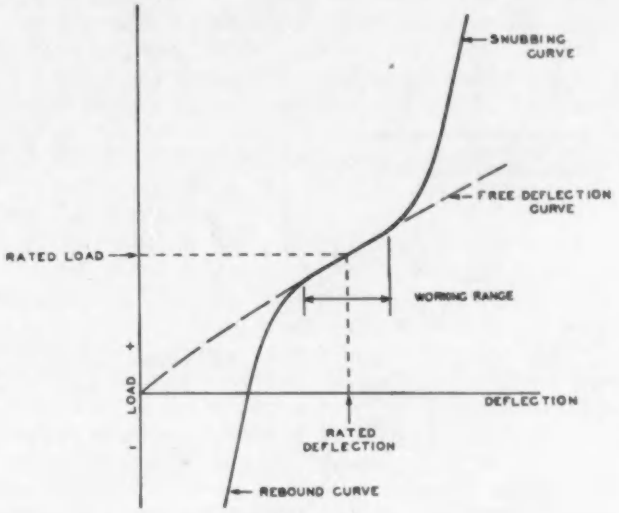
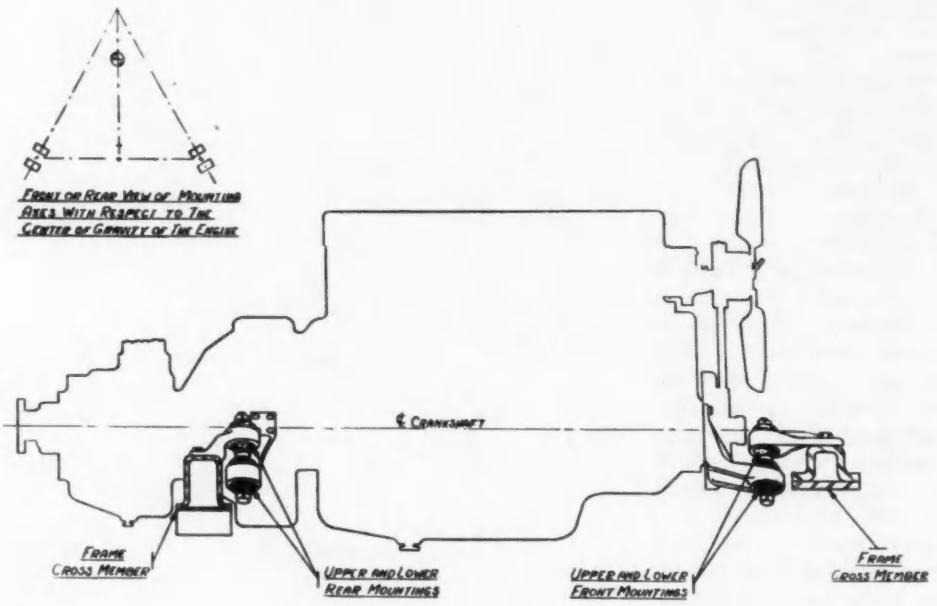


Fig. 4 – Load-deflection characteristic for a shear mounting with provision for snubbing against shocks

passing through the individual centers of gravity of the engine and transmission; this arrangement will normally position front mountings at a higher elevation than at the rear. By utilizing the "ideal" set-up, translational vibrations produce pure translational movement of the engine and torsional vibrations produce only torsional movement. Without this mounting arrangement translational and torsional effects become "coupled," causing more complex engine movement, confusing natural frequencies, and detracting from stability. Moreover, mountings are commonly designed and installed to be softest along their vertical axes, and comparatively stiff in other directions, as a result of which a pure vertical character of deflection for both translational and torsional effects is normally most effective, which is the ease with an "ideal mounting" installation, with all mountings positioned upright in the same plane as the "principal axis."

Another phrase for "ideal mounting" is "center-of-gravity support," expressing that the mountings are in effect positioned at the center of gravity and supporting the load at that point. By the same token a "virtual center-of-gravity suspension" is an arrangement in which mountings positioned remotely from the center of gravity are so engineered as to offer the same results as a center-of-gravity support. This arrangement entails proper consideration of the angle of positioning of the mountings, their location, and the ratio between vertical and horizontal stiffnesses. Such a

system can be looked upon in effect as lending a "track" to guide the character of movement of the powerplant. In any such arrangement it is desirable to locate the forward universal joint of the propeller shaft in the "ideal mounting" plane so that its displacement is excited by only chassis movement and not that of the powerplant due to usually favorable though it may encourage resonance in cases where a nearly critical condition exists between engine vibrations and the mountings proper. In aircraft work this structural stiffness is computed carefully and used as a correction factor for theoretical mounting stiffness.



■ Fig. 5 – Virtual center-of-gravity suspension

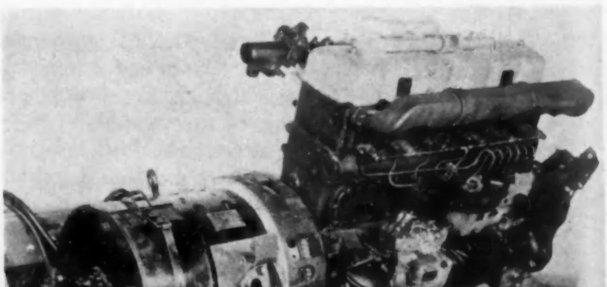
vibration. Fig. 5 illustrates a virtual center-of-gravity support; the arrangement shown renders the advantages of being mounted apparently at the center of gravity and also has an extremely low rotational stiffness for remarkable torsional cushioning and consequent smoothness at low driving speeds (due to the use of a pair of mountings in series at each point of support).

"Three"- and "four"-point supports are common arrangements. A three-point suspension is one in which engine torque is restrained by two mountings only, usually those at the flywheel housing or transmission positions; a trunnion mounting at the fan end of the engine spanning mounts at both sides of the frame is still known as a three-point suspension in that no torque is carried by the points at the trunnion. The arguments in favor of three-point mounting are: (a) it is impossible to set up a strain in the engine due to chassis distortion; and (b) the torsional natural frequency is very low, giving a high degree of isolation. The four-point arrangement, resisting torque at all four points of mounting, lends remarkable stability to the powerplant but, due to its greater torsional stiffness, it does not usually render as effective torsional vibration isolation as a three-point mounting. Fig. 6 is a photograph showing the powerplant of a diesel-electric bus with its rubber mounting installation in which a four-point arrangement is used.

Naturally it is true that the aforementioned considerations theoretically deal with a powerplant mounted to a rigid structure through soft mountings. Actually there is appreciable resilience to a vehicle body which contributes to the overall spring rate of the suspension; this effect is

It may be recognized readily from what has been said that step No. 1 in laying out an engine rubber suspension is to decide upon the proper stiffness of mountings (that is, static deflection) to be used, by considering engine characteristics and speeds of operation. Next, these units should be chosen to suit the loads supported at the points of mounting. Having provided for these fundamentals, the other details will depend upon the conditions of the particular installation.

Typical cases of mounting "headaches" are those re-



■ Fig. 6 – Powerplant of diesel-electric bus with four-point arrangement of rubber mounting installation

sulting where slight revisions of conditions have been made over a period of time without detriment, following which some one minor change will "break the camel's back." Such was the case when a very unpleasant throbbing sensation developed at idling speed in a motorcoach wherein the engine mounting conditions had not been altered from those which had prevailed favorably for several model years. At length it was discovered that resonance existed between translational vibrations set up by the engine and the "organ-pipe" natural frequency of the coach body, the length of which had just been increased slightly. The obvious way to avoid this condition was to lower the idling speed but, in so doing, resonance developed between engine translational effects and rubber-mounting natural frequency. This indicated the necessity of using mountings of greater deflection to lower their natural frequency below the new idling speed.

Beside this effect and the fluid-drive stall-speed consideration mentioned earlier, many other problems have had to be worked out in modern rear-engine motorcoaches. This experience should stand us in good stead in the handling of similar situations in passenger cars of types anticipated in the coming years. For one thing, in "cross-coach" rear-engine installations, there is a distinct tendency toward rocking of the powerplant under clutch engagement and braking; this is best combated by carefully designing toward a center-of-gravity suspension so that translational forces of external origin will not produce rotational effects of the engine relative to the structure. Moreover, rear-engine coaches, with engine either across or along the main axis of the vehicle, are further faced with a condition wherein movement set up by the engine is modified in any levers or controls running the length of the coach. Unless care is taken by proper engine mounting to keep movement to a minimum, chattering of controls at the forward end may become extreme, even to the extent of throwing the job out of gear on rough-road operation.

Beside noise and vibration isolation and stability in the presence of external shocks, the types of protection rendered by a proper shear-loaded rubber mounting system are: (a) an appreciable degree of cushioning of the mounted equipment with respect to external impacts, such as shellfire experienced by military vehicles (either in the case of guns mounted on the vehicle or shells hitting the vehicle); and (b) ability to tolerate twist of the chassis without straining engine members.

There is no question that rubber mounting engineers would be in a far better position to assist the equipment designers if the attitude: "Here's a powerplant, how will we mount it?" were entirely superseded by: "We are laying out an engine design. What provision will we make for rubber mountings?"

This discussion has revolved around the main factors that are necessary to consider for good engine mounting procedure. There are other detailed considerations entering into each installation which tie in with the characteristics of that particular assembly and are accordingly beyond the realm of the general treatment brought out herein. The progress in the past several years, brought about by recognizing the variables involved and through the close cooperation of the many interested parties, has been very noticeable and, as the principles of vibration isolation are used to a greater extent in the future, all types of motor vehicles will become increasingly smoother.

Development of Impact-Resistant Windshields

AT this time when the nation's air transportation system is important to the war effort, the maintenance of safe and reliable operation assumes added significance. Accordingly, the conduct of technical development looking to the minimization of causes of accidents not only is necessary as a safety measure but has become a defense urgency.

One of the most serious potential hazards to air-carrier operation involves the existing lack of means for adequately protecting windshields against collision with birds during flight. Such collisions usually result in the immediate and total decease of the birds. Unfortunately, however, they also can result in the destruction of the airplanes involved and in the death of their occupants.

Impact forces in such collisions are enormous. Even small birds such as ducks not only have penetrated the windshield, but one in particular continued through the bulkhead, traveled the length of the cabin, penetrated the rear cabin wall, and lodged finally in the baggage compartment. Fortunately in this case neither the passengers nor the crew were struck.

The possible destruction and loss of life resulting from such a collision with a duck is unpleasant to contemplate. However, the force of such an impact is multiplied several times when an airplane disputes the right of way with a swan. This has occurred. In fact the collision in question involved five swans. The pilot of that airplane reported as follows:

"Time 12:17 a.m. – Climbing at 8000 ft – Air speed 150 mph – Hit flock of swans – One swan penetrated leading edge, left wing – Second swan almost tore off left vertical stabilizer – Rudders jammed – Third swan struck and dented engine cowl – Later, two swans went through propeller – Portion of swan taken from wing after landing, weighed 11½ lb."

Note: Wild swans weigh as much as 20 lb.

Accidents involving bird collision have become alarmingly numerous. A partial record, obtained by M. Gould Beard of American Airlines, includes 61 such collisions since 1939, two-thirds of which occurred at night and more than one-third of which resulted in the penetration and shattering of the windshield.

The development of means to protect aircraft windshields from such collisions appears to the Technical Development Division of the Civil Aeronautics Administration to be a "must" project. This urgency further has been emphasized by numerous requests from the industry and from other bodies within the Administration that such development be undertaken. This now is in process.

The prospect of stopping a 20-lb swan at a relative approaching velocity of 270 mph at first was somewhat

disheartening, particularly since de-icing means and protection against visual and acoustical shock due to lightning necessarily are involved in any windshield development problem. Continued studies and investigations, however, have indicated that there is considerable promise of providing a large degree of protection, and that such protection need not involve excessive weight or complication.

Thus, among other possibilities, the use of a retractable metal screen or similar arrangement was considered, although the mechanical difficulties involved made its practical realization somewhat doubtful. Recently, however, developments of transparent glass-plastic combinations have resulted in windshield panels of reasonable thickness that have proved highly resistant to impacts, as indicated by numerous pressure and impact tests and by computations based upon those tests.

Those pressure and impact tests were carried out by the manufacturer of the glass-plastic panels, by Dr. F. W. Adams of the Mellon Institute of Industrial Research, by Dr. G. M. Klein of the National Bureau of Standards, and by personnel of the Civil Aeronautics Administration. Pressure tests of the glass-plastic windshield panels indicate that static pressures of at least 35 psi can be withstood by a 14 x 28 in. panel ⅝ in. thick, and that the panel will undergo large deflections of several inches magnitude before the plastic layer will rupture. Dr. Klein's tests on the recently developed glass-plastic windshield panels further indicate that at normal temperature those panels are far more resistant to penetration by a falling dart than is the conventional type of windshield.

The tests carried out by Dr. Adams to determine qualitatively the effect of variations in the consistency of a projectile striking a glass panel revealed secondary effects caused by the velocity of the projectile, and the effect of variations in angle of incidence during collision. It was indicated that a semi-liquid projectile, such as a bird carcass, has considerably less penetrating power than a tough rubber-like projectile, that at high velocities this effect becomes even more pronounced, and that the resistance of a windshield at 45-deg incidence to the path of a projectile is much greater compared to resistance at normal impact than would be expected from geometric theory.

■ Results of Early Tests

A further series of tests was made by the Civil Aeronautics Administration in which a compressed-air gun with a 3½-in. diameter barrel and operated at 200 psi pressure was utilized to project freshly killed chickens against a backstop. It was learned from such tests that the chicken carcass would be completely flattened and shredded by a 200 fps impact, although still hanging together in one mass, and would cover an area of approximately 100 sq in. on the backstop. It further was indicated by those tests that it is practical to utilize a freshly killed bird carcass for test purposes.

The degree of protection provided either by a retracting shield arrangement or by improved windshield materials appears doubtful until tests which simulate actual flight conditions can be conducted. The Technical Development Division, therefore, has considered that as a prerequisite to obtaining adequate windshield protection, it is essential that a satisfactory testing method be developed: for obtaining fundamental design data; for evaluating the degree of protection afforded by presently existing windshield materials; to provide a basis for indicating when adequate protection finally is obtained; and for determining the resistance of other portions of aircraft structures against such impacts. It has appeared evident in such connection that the testing method should utilize a catapult to project a simulated or actual bird carcass against the test structure at a velocity equal to the velocity at which the bird and airplane approach each other under flight conditions. It furthermore has been indicated through considerable investigation that the most practical type of test catapult is a compressed-air gun.

In order to obtain preliminary data for the design of such a gun and further to determine the practicability of its use, as described before, a series of tests was made with a small gun available at the National Bureau of Standards from which freshly killed chickens were shot. As previously mentioned, it was concluded from those tests that a bird carcass can be propelled from such a gun without appreciable damage to its body and that the complete flattening and spreading which occurs upon impact would be difficult or impossible to simulate with anything but an actual bird carcass.

■ Larger Gun Being Developed

As a result of those experiments it is considered desirable to attempt the development of a larger air gun which would be capable of projecting 16 lb of de-winged swan carcass at a velocity of 270 mph. Accordingly, arrangements are being made by the Civil Aeronautics Administration to negotiate a contract with the Westinghouse Electric and Mfg. Co., to provide a complete test set-up at its East Pittsburgh plant. This set-up will include a compressed-air gun using either of two 20-ft barrels of 5-in. and 10-in. diameters, means for mounting the forward cabin or other portion of airplanes in front of the gun, and means for measuring the velocity of the bird carcasses as they are projected. In addition, it is planned to obtain high-speed motion pictures during the tests and to install strain gages and accelerometer pick-ups at various points on the structure in order to obtain impact forces, stresses, and other data on the windshields and supporting structures.

In addition to its use by the Civil Aeronautics Administration, it is planned to make the test set-up available to other designated organizations so that windshields, windshield protecting devices, or other portions of airplanes may be tested from time to time as may prove desirable. However, since a number of recently developed windshield panels now are available, it is felt that the first series of tests should involve the forward cabin portion of an air-carrier type airplane with various panels installed.

Attempts now are being made to obtain such a structure. The Pittsburgh Plate Glass Co., and the Libby-Owens Ford Glass Co., have indicated their desire to submit test panels. F. C. Lincoln of the Division of Wild Life Research of the Department of the Interior has been most helpful in supplying information concerning the weights and proportions of migrating birds and is aiding in our efforts to obtain bird carcasses for testing purposes. It is hoped that members of the industry and interested Government agencies will participate in this program in every way possible.

Excerpts from the paper of the same title by A. L. Morse, Civil Aeronautics Administration, presented at the National Aeronautic Meeting of the Society, New York, N. Y., March 12, 1942.

FUEL FEED at HIGH ALTITUDE

by W. H. CURTIS and R. R. CURTIS

Thompson Products, Inc.

THE greatly accelerated rate at which designs of military airplanes with increased performance have been developed has added materially to the difficulty of feeding vapor-free fuel to the carburetors at the higher altitudes. The fact that vapor lock is now being encountered in simple fuel systems far below the altitude where it would be predicted from considerations of the temperature and pressure within the system at which air-free boiling would begin, has made the problem extremely acute in some instances. Moreover, the remedy, where the airplanes were already in production, has been costly both in its determination and application. This situation has resulted in the initiation of a number of independent investigations, no one of which appears to be correlated to the others. Nearly all are aimed at the development of data by ground tests that will reliably indicate the performance of the airplane's fuel system at high altitude.

The research that led to the development of this material also can be said to be one of several, but it is probably the oldest continuous study that has made use of well-developed ground or laboratory equipment. It has been frankly independent and uncorrelated with any other similar activity. The opinions and data are therefore presented largely as our own. However, much of the research has now been reduced to practice and, to this extent, proved. It is therefore hoped that this discussion may serve as an aid in setting up standard procedures for such work and also that it may be helpful to those who find it necessary to undertake fuel-system studies for themselves.

Adequate fuel feeding at altitude is a matter of vapor elimination, either by preventing its formation or by removing it from the system in the event that its formation cannot be prevented. This vapor, as it appears in fuel systems, may consist of atmospheric air saturated with fuel vapors, or it may consist mainly of the gaseous product of fuel boiling, depending upon the temperature and absolute pressure to which the fuel is subjected. But, regardless of

ADEQUATE fuel feeding at altitude, these authors point out, is a matter of vapor elimination, either by preventing its formation or by removing it from the system in the event that its formation cannot be prevented. The effect of vapor is invariably to cause failure of the fuel flow if it forms in sufficient quantity in any part of the fuel system that lies between the fuel tank and the carburetor.

This paper gives the results of a study of the conditions that bring about this type of fuel failure, and describes means of exploring the phenomena experimentally so that it can be ascertained in advance of manufacture if remedial steps are necessary.

The greatly accelerated rate at which designs of military aircraft with increased performance have been developed, they explain, has added materially to the difficulty of feeding vapor-free fuel to the carburetors at the higher altitudes.

The influence of the following variables that affect aviation fuel during flight to high altitude is discussed: dissolved air, vapor pressure, fuel temperature, turbulence, velocity of fuel flow, rate of climb, altitude, vent-line effects, and heat transfer.

In the first part of the paper the simulation equipment is described and illustrated; simulation test procedure is specified; and experimental observations are discussed. The second section, on the centrifugal booster pump, gives the results of three series of tests: one on an isolated tank of fuel, and two on the booster fuel system.

* * *

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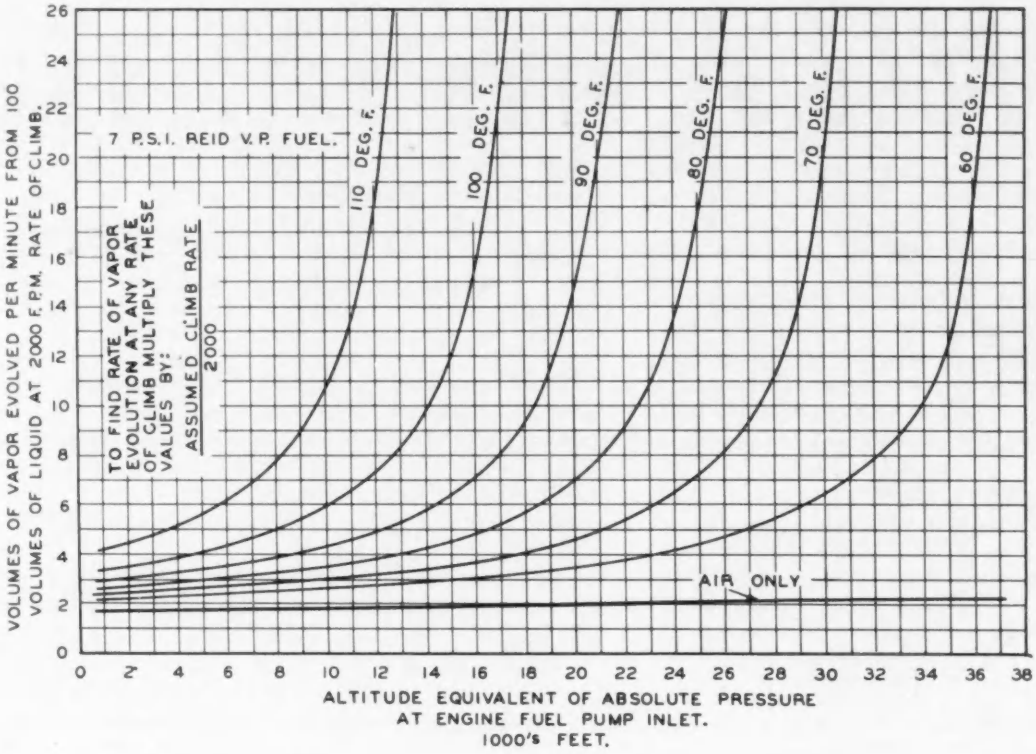
[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Oct. 30, 1941.]



■ Fig. 1 – Curves for predicting the critical altitude for simple fuel systems for fuel having a Reid vapor-pressure of 7 psi (assuming suitable compensation for effect of air evolution lag)

its composition, its effect is invariably to cause failure of the fuel flow if it forms in sufficient quantity in any part of the fuel system that lies between the fuel tank and the carburetor. Our concern then is not only to learn as much as we can about the conditions that bring on this type of fuel failure, but to develop convenient and acceptable means of exploring the phenomena experimentally so that we shall be able to ascertain in advance of manufacture if remedial steps are necessary and to evaluate them in the event it is determined that revision is required.

■ Terminology

As here used, *fuel-flow failure, fuel-system failure,* or *vapor lock,* does not refer to total loss of fuel pressure. Rather, it is predicated on the amount of reduction in fuel pressure at the carburetor that will cause the latter to cease functioning properly. In practice, a drop of 2 psi is often as much as can be tolerated. In experimental work, such as we are concerned with here, a drop of 1 psi is sometimes considered unsatisfactory.

The term *simple fuel system* is intended to indicate a system of fuel tanks with outlet lines connected through a tank selector cock to a fuel pump mounted on the engine. The presence of an engine selector cock, strainer, or a hand pump in the fuel line between the tank selector and the fuel pump on the engine does not alter the system. It is not intended to include systems incorporating a booster pump or other aid designed to force fuel from fuel tank to fuel pump.

Fuel pump refers only to the fuel pump that is mounted on and driven by the engine. In Section II it is referred to as *engine pump.*

Booster pump or *booster* refers to any power-driven pump installed at the fuel tank for the purpose of assisting the fuel pump in its duty of pumping fuel at high altitudes.

Effective altitude refers to the altitude equivalent of the lowest absolute pressure within the fuel system. In simple fuel systems this pressure usually exists at the fuel-pump inlet.

I – Fuel-System Variables and Simulation

■ Variables During Climb

When an airplane is flown from the ground to high altitude at a rapid rate of climb, the most critical factors affecting the fuel are the rate and the amount of atmospheric pressure drop within the fuel tank. This is true regardless of the presence or absence, at the take-off, of equilibrium between the fuel and its vapor. What occurs, as the result of this pressure drop, depends upon a number of variables in both the fuel system and the fuel, most of which are well known. However, their influence upon vapor lock during this period cannot yet be fully evaluated. The following variables appear to be important:

1. Dissolved air.
2. Vapor pressure (of the fuel) and its corollary – air-free initial boiling point.
3. Fuel temperature.
4. Turbulence.
5. Velocity of fuel flow (in fuel lines) and its corollary – minimum absolute pressure (within the system at any instant).
6. Rate of climb.
7. Altitude, or its corollary – absolute pressure on the fuel in the fuel tank.
8. Vent line effects, which may modify (6) and (7).
9. Heat transfer (from fuel to atmosphere).

Most of these variables are dependent upon one or more of the others, and some are so closely related as to make it difficult to discuss them alone. In the limited discussion that follows, some will, therefore, be treated as related

groups, each being headed by a single variable that appears to be influenced by all others included in the group.

Dissolved Air – The behavior of dissolved air is influenced by all of the eight other variables listed.

Aviation fuel, when placed in a vented container, will release or dissolve air until the sum of the partial pressures of vapor and air within the container equals the pressure of the ambient atmosphere. Thus it follows that there will be a decrease in the amount of dissolved air with an increase of fuel vapor pressure or with a decrease in ambient air pressure. Also, since fuel vapor pressure increases with temperature rise, the latter will cause a decrease in dissolved air.

The rate at which the temperature or pressure is changed influences the stability of the air-in-liquid solution. With quiescent fuel, if temperature rise or pressure drop, or both, occur at a sufficient rate, the solution will become unstable in that it will be supersaturated with air. When this condition exists, excessive amounts of air will be released at points where sharp hydraulic pressure drop or turbulence occurs. Or, with continued quiescence, the evolution of the air will lag, in which case it will persist for some time after the terminal temperature and pressure have been reached. This lag is reduced, and may disappear altogether, if sufficient turbulence is present.

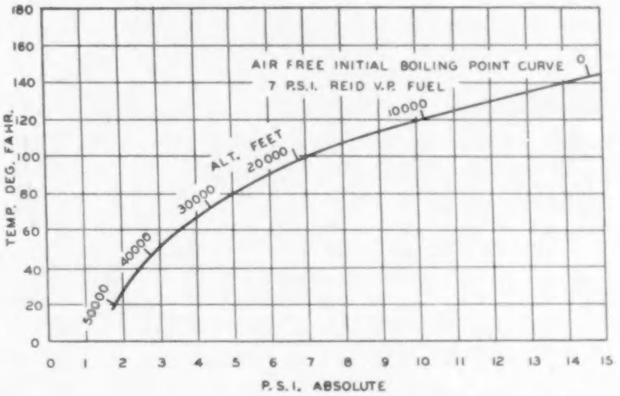
Since a reduction in the air pressure above the fuel will result in a decrease of the amount of dissolved air within the fuel, it follows that air is evolved from the liquid throughout all parts of the fuel system wherein a drop in absolute pressures occurs during the climb of an airplane from the time it leaves the ground. The rate at which the air is released from solution is a function of the rate of climb and turbulence. Theoretically, the rate is unaffected by fuel temperature, notwithstanding the greater air content of colder fuel. This may be accounted for by the higher altitude to which colder fuel must be taken before it will boil and the greater time thus allowed for the escape of the dissolved air. Under the theoretical condition of a very high rate of climb with no turbulence, it appears likely that the evolution of the air would lag at altitudes below that at which air free boiling begins and that, at this point, the resultant supercharge of air would be released within a very short space of time, for there can be no air in the fuel once true boiling is well under way. However, there has been no indication to date that this condition can exist to any appreciable degree in an airplane fuel system. It may be present to some extent in simulation equipment, by reason of the absence of vibration.

While air is evolved during a decrease of pressure in the fuel tank and in the fuel lines that extend therefrom to the fuel pump, that which is released in the fuel tank appears to do no harm. In the fuel lines, however, trouble quickly develops. Here, there is usually great turbulence accompanied by lower pressures than in the fuel tank, with the further unfavorable condition that the lowest pressure is at the fuel-pump inlet or even within the pump itself. These conditions usually result in an increasing percentage of air making its appearance as the fuel progresses from the fuel tank to the pump. If we were dealing with air alone, this would not be a serious matter, for there appears to be insufficient air present in aviation fuel to cause vapor lock. But, as the air is evolved from solution, fuel vapors accompany it in increasing proportions as the air pressure over the fuel decreases. By reference to Fig. 1, it will be seen that the rate at which air is released from the fuel is nearly constant throughout any constant-rate climb, but that the amount of fuel vapor carried along with it increases rapidly some time before the boiling point of the fuel is reached. This effect is so great, in fact, that vapor lock generally occurs well before true boiling of the fuel. For example, consider the curve for 100-F fuel, which reaches its air-free boiling point at an altitude of slightly less than 20,000 ft. Experimental and other data of a limited nature lead us to suspect that vapor lock may be encountered in simple fuel systems at a rate of vapor evolution as low as 8 volumes per min at 2000 fpm climb, which would indicate a critical altitude of nearly 13,000 ft. Also, this is the equivalent altitude as determined by the absolute pressure at the fuel pump inlet and, since the pressure at this point is usually less than that of the ambient air, the airplane itself will generally be at a still lower altitude. In some cases, this difference is as much as 3000 ft. Thus it becomes evident that it is possible to experience vapor lock at an altitude 10,000 ft below that at which true, or air-free, boiling of the fuel will make itself evident.

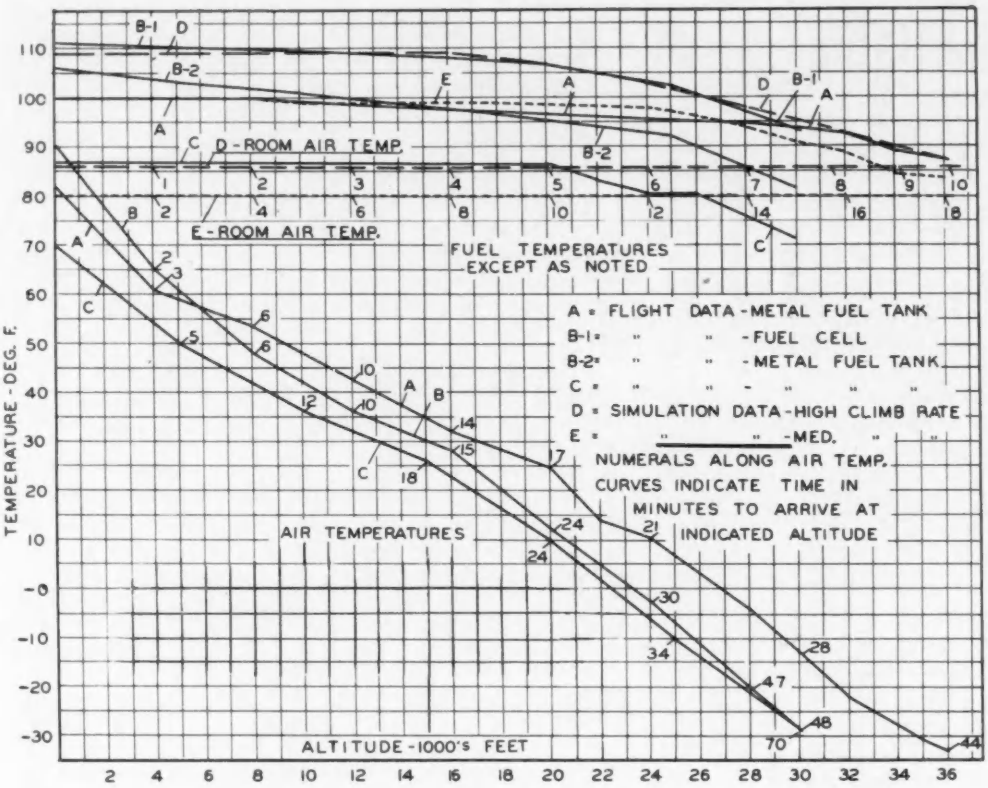
The values used in plotting the several curves in Fig. 1 have been calculated by means of the well-known partial-pressure equations. The graph is for a fuel having a Reid vapor pressure of 7 psi, and is purely theoretical. However, this method appears to hold some promise of affording a convenient and reasonably accurate means of predicting the critical altitude for simple fuel systems if suitable factors for giving effect to air evolution lag can be found. There appears to be no other fundamental variable that would seriously impair its accuracy.

Vapor Pressure – The vapor pressure of the fuel depends upon its composition and temperature. It is not a direct indication of composition, however, for composition can vary considerably with different gasolines, all of which may have the same vapor pressure. Vapor pressure, moreover, will rise with increase in temperature until it has the same value as the pressure of the atmosphere and, at this point, if equilibrium conditions have been maintained during the temperature rise, boiling will begin. A curve of these initial boiling points is shown in Fig. 2. This curve indicates the temperatures and absolute pressures at which an air-free fuel with a 7 psi Reid vapor pressure may be expected to start boiling.

The indicated vapor pressure of any given fuel will vary according to the procedure used in determining it. The Reid method is commonly used and its values are always



■ **Fig. 2 – Temperatures and absolute pressures at which an air-free fuel with a 7 psi Reid vapor pressure may be expected to start boiling**



■ Fig. 3 – Comparative observed fuel and air temperatures during flight to altitude and typical simulations

taken at 100 F. In technical work, air-free vapor pressure is frequently used for the reason that it is considered as reliably indicating true air-free boiling points. At 100 F, the air-free vapor pressure is obtained by multiplying the Reid vapor pressure by 1.03.

Fuel Temperature – Fuel temperature assumes importance chiefly by reason of its effect upon vapor evolution. Fig. 1 illustrates this point clearly. It is an ever-changing variable, almost completely uncontrolled in the airplane and therefore difficult to duplicate in simulation equipment. It is, of course, affected by the air temperature when fuel is held above ground in either the airplane or mobile fueling equipment for any length of time. Moreover, it may, if the equipment in which it is stored is exposed to direct sunlight, assume a temperature well above that of the surrounding air. Airplane fuel tank temperatures of 115 F have been observed under this condition when the official Weather Bureau temperature did not exceed 90 F.

High altitude, with its concurrent low air temperatures and pressures, appears to have little effect on fuel temperature until the pressure is reduced to the point at which true boiling starts. A pronounced temperature drop then begins. By referring to Fig. 3, it will be observed that the amount of this drop is less influenced by rate of climb than by altitude. Comparison of the simulation and flight data shown in this graph leads to the conclusion that most of the fuel temperature drop in either simulation or flight is due to the extraction of heat from the liquid to support boiling.

All of the data for plotting the curves in Fig. 3 were taken by experienced engineers with adequate equipment; yet it will be noted that there is marked disagreement in some of the flight temperature data. Flight *A*, for instance, shows a fuel temperature drop of only 13 F, whereas Flight *B*-2 developed a drop of 23 F, notwithstanding the fact that Flight *A* was to a higher altitude. No satisfactory explanation for this apparent discrepancy has been found. There is great need for more flight data of unquestionable accuracy.

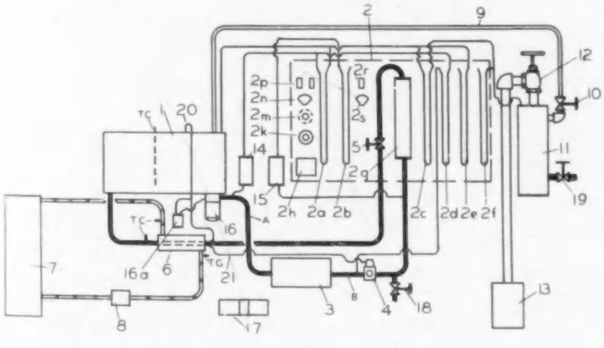
Turbulence – There is reason to believe that turbulence is both beneficial and harmful to proper fuel-system functioning, according to where it occurs. That it is present throughout the system appears to be beyond question. Theoretically, it should assist in the maintenance of a stable liquid during the climb to altitude. If this is true, it must be an accessory to the evolution of dissolved air during the pressure reduction that accompanies climb. This action in the fuel tank, it can be reasoned, will be helpful by reason of the fact that air released at this point leaves the system through the vents. However, in the fuel lines, the released air must pass through the fuel pump where, with its accompaniment of fuel vapor, it can and usually does cause serious fuel pressure drop.

Velocity of Fuel Flow – Velocity of fuel is a controllable variable inasmuch as it depends upon the engine demand and the size of the fuel lines. Some effort has been made to develop data that would indicate the most favorable size of fuel line to use for a given fuel demand, but the results have been inconclusive. This condition is due, in major part, to the peculiar accumulative effect of vapor in the lines. Once it begins to form, its throttling effect serves to increase both the velocity of flow and the turbulence and to decrease absolute pressure, whereupon additional vapor is released and the process of vapor lock is well under way. This behavior is favored where lines with unfavorable slopes are used or where humps exist that can form vapor traps. Smaller lines, with higher flow velocities, will prevent much of the vapor trapping but, on the other hand, the greater pressure drop in small lines may, of itself, be sufficient to cause vapor lock.

Rate of Climb – The rate of climb, the altitude, or the absolute pressure on the fuel in the fuel tank and the vent line effects are so closely related that it is impossible to discuss them separately. The rate of climb certainly governs the rate at which the absolute pressure within the fuel system can fall, while the conditions in the vent line have much to do with the rate at which it actually falls. Ram effects on the vent line can be, and are, used to establish pressures within the fuel tank above that of the ambient atmosphere. This, unquestionably, has a beneficial effect upon fuel system performance, for the reason that it lowers the effective altitude within the fuel system. As a result, the rate of vapor evolution is lower. The influence of this depressed effective altitude is obvious from a study of Fig. 1.

■ Simulation Equipment

It is not practical to conduct primary fuel-system research in the airplane for it is impossible to arrange adequate instrumentation and control, and it is equally impossible to provide the necessary points in fuel lines and tanks for visual observation. Moreover, experimentation with the airplane is prohibitory in cost. Still another point against it is the fact that no hint of trouble is evident until the airplane can be built and flown, whereupon any required remedy usually involves costly changes. It is also possible to obtain a solution in the airplane, with insufficient reliable information regarding the reason for it, whereupon little is gained that may be useful in future design.



■ Fig. 4 – Diagrammatic sketch of simulation equipment

Fig. 4 shows, in diagrammatic form, an arrangement of simulation equipment which at the present time seems to do the best job, although it is to be admitted that the test procedure has not been perfected to the point where it exactly simulates the fuel system in the airplane. In general, however, any error is on the safe side. Two views of one arrangement of this equipment are shown in Fig. 4A.

The reference numbers in Fig. 4 indicate the following:

1 – Fuel tank – capacity with 10 in. of fuel, approximately 20 gal.

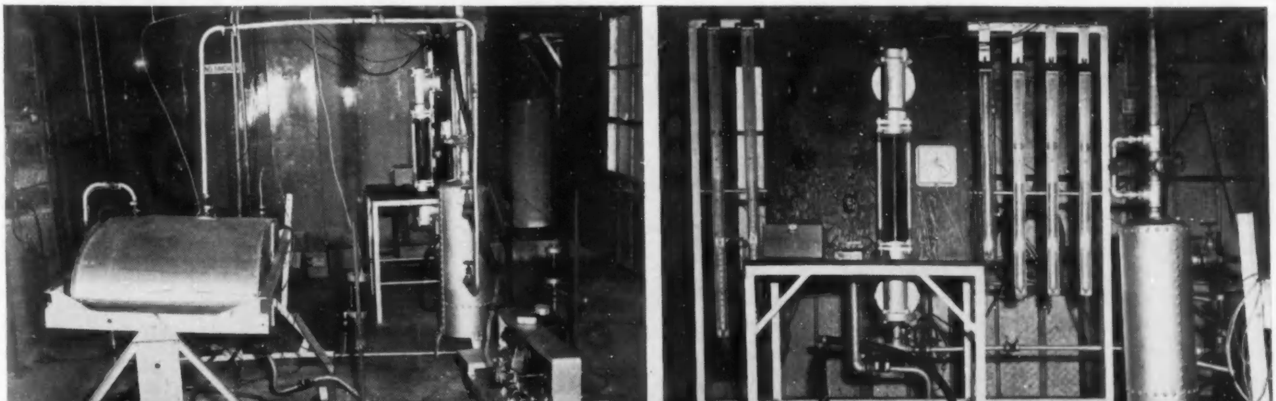
2 – Control panel. Minimum equipment, as follows:

- 2a – Manometer for booster-pump discharge pressure.
- 2b – Manometer for fuel-pump discharge pressure.
- 2c – Manometer to read directly the difference in pressure between the altitude tank and the fuel tank.
- 2d – Manometer to read pressure drop in system between booster pump and fuel pump.
- 2e – Manometer, graduated in altitude, to indicate directly the altitude inside the fuel tank.
- 2f – Manometer, graduated in altitude, to indicate directly the altitude of the airplane as determined by the vacuum in the altitude tank.
- 2g – Flowmeter.
- 2h – Potentiometer.
- 2k – Potentiometer switch.
- 2m – Voltage control for motor-generator set.
- 2n – D-c voltmeter – 30-v scale.
- 2p – Switches for 12- or 24-v motors on boosters.
- 2r – Switch for fuel-pump motor.
- 2s – D-c ammeter – 30-amp scale.

TC – Essential points for thermocouples for temperature readings.

3 – Fuel panel – a mockup of all selector valves, line strainer, and emergency fuel pump, if one is used at this point. (Sometimes omitted where fuel-line conditions are favorable with respect to pressure drop.)

4 – Fuel pump – here driven by an electric motor through a variable-speed drive. Pump rpm should be adjustable over a minimum range of 1200 to 2700 rpm.



■ Fig. 4A – Two views of one arrangement of simulation equipment

5 – Valve for controlling rate of fuel flow. Note that this is on the discharge side of the flowmeter.

6 – Heat exchanger for heating fuel. May require a fuel bypass if high in residual heat capacity.

7 – Hot-water heater.

8 – Circulating pump for hot water.

9 – Vent line.

10 – Vent control valve – for maintaining ram or tank pressure values which may be read directly on manometer (2c).

11 – Altitude tank. So called because the pressure within it represents that of the ambient atmosphere at the altitude of the airplane.

12 – Vacuum control valve to regulate rate of climb.

13 – Vacuum pump – preferably a commercial type of not less than 100 cfm displacement and with a terminal pressure of 1 mm hg or better.

14 – Manometer well for booster pump discharge pressure – eliminates static head errors on 2a and 2d. May also be required on other leg of manometer 2d.

15 – Manometer well for fuel-pump discharge pressure – same function as (14), except used on 2b only.

16 – Booster pump – may be in different location on fuel tank than that shown. Must be positioned exactly as called for in the airplane fuel system design.

16a – Glass chamber for liquid trap.

17 – Motor-generator set for furnishing current to any 12- or 24-v motor such as that on the booster or an electric motor-driven emergency fuel pump.

18 – Drain valve.

19 – Vacuum relief valve.

20 – Seal balancing line to booster pump seal drain chamber. To prevent unseating of seal when vacuum is imposed within the fuel tank.

21 – Relief valve balancing line to fuel pump. Used rarely and only in the event that it is necessary to simulate non-turbo-supercharged operation.

Electric wiring, fuel storage, and filling arrangements are obvious.

Transparent sections for visual observation are usually provided at points A and B in the fuel line. It is also desirable, but not always possible, to have windows in the fuel tank to permit observation of the booster pump action where this type of equipment is being used. Mounting the fuel tank in a cradle that is arranged to permit tilting it at any angle, or even rotating it through a complete circle on at least one axis, is advantageous, particularly in reducing the labor of setting up and altering test arrangements.

■ Simulation Test Procedure

All reference numbers, unless otherwise indicated, appear in Fig. 4.

Fuel – Only fresh fuel that has not previously been subjected to altitude tests should be used. Reasonable precaution against error requires that it be of the same grade as that which is to be used in the aircraft fuel system that is being simulated. A Reid vapor pressure test should be run at intervals sufficiently frequent to detect weathering or change of composition before it has progressed to the point where it can affect the reliability of the tests. The Reid vapor pressure appears to be a more convenient and reliable index for this purpose than an ASTM distillation. With proper equipment, it is also more quickly obtained.

Heating of the Fuel – It is considered essential that the airplane be capable of taking off and flying forthwith to high altitude with warm fuel in the tanks and, for this reason, it has become standard procedure to run all simulations with fuel at an elevated temperature. The tentative standard temperature at the start of the simulation, or at take-off for the airplane, is 110 F, which is much higher than temperatures usually reached in storage. It therefore becomes necessary to provide suitable means for heating the fuel. It has been reasonably satisfactory to do this by circulating it through a heat exchanger wherein the heating medium was circulating hot water, although it is to be admitted that this method fails to simulate the natural heating that may occur when either the airplane or above-ground fueling equipment is exposed to the sun. The heating in this case is very slow and, for this reason, ample time exists for complete stabilization of the fuel. To hold costs and time within reasonable limits, the heating of the fuel in the simulation equipment must be much more rapid than natural heating; however, there is reason to believe that, when this heating is accompanied by circulation and some turbulence, reasonable stability is attained at the terminal temperature. It follows, of course, that, even with such turbulence as may be set up in a closed circulation circuit, the heating must not be too rapid and the test must not be started until the final temperature has stabilized. Also, the vent must be open to the atmosphere during all of this time.

Circulation of the fuel during the heating is best accomplished by the fuel pump (4). At least four thermocouples should be used to aid in the control of heating rates and final temperature. One of these should be located at the water inlet and another at the water outlet of the heat exchanger. One is located at the fuel outlet of the heat exchanger and one should be located well below the surface of the fuel in the fuel tank. No standard for the rate of heating has been adopted.

It has been our practice not to add heat during the simulation test notwithstanding the greater fuel temperature drop observed in simulations over that reported in airplane flights. Apparently, there is a greater heat loss by radiation in simulation equipment. It has also been reasoned that there is more boiling of the fuel than in the airplane, but no data other than the relative temperature drop have been offered to support this theory and, quite recently, some carefully conducted fuel-temperature studies in a large bomber disclosed as great a temperature drop as any that has been observed in simulation equipment.

Vent Line Effects – The vent line (9) should either simulate the aircraft vent line, or the valve (10) should be used to impose any ram or pressure values within the fuel tank that have been calculated or determined by test flight. If the outlet of the vent line is so located on the aircraft that neither ram nor suction results, then the vent line may be simulated by changing the length only in accordance with the equation:

$$L_1 = L\left(\frac{V}{V_1}\right)^2$$

Where:

L = Length of vent line in aircraft, ft.

L_1 = Length of vent line in simulation test, ft.

V = Volume of fuel in aircraft fuel tank to same depth as that in the simulation equipment.

V_1 = Volume of fuel in the simulation system fuel tank.

Thus it will be seen that, with one-half the volume of fuel in the simulation fuel tank, the vent line for the conditions noted should be four times as long as that in the aircraft, provided it is the same in diameter.

It is not considered possible to obtain vent-line simulations under all conditions. In general, it probably will be more satisfactory to control the fuel tank pressure to a known or calculated ram or pressure value. Thus, if it has been determined that a minimum ram of 13.5 in. of water pressure is to be maintained under the flight conditions being simulated, valve (10) would be manipulated during the test to cause manometer (2c) to show 1 in. hg higher pressure in fuel tank (1) than in altitude tank (11). This presupposes, of course, that the vent arrangement in the airplane is such that the back pressure of escaping vapor is negligible. And, in the main, this is a reasonably safe assumption unless the vent line is unusually small. Therefore, if the ram pressure has been determined by the designing engineer, it seems best to work from this basis rather than to attempt vent-line simulations. Moreover, the ram pressure has now become so important that, in most cases, vent-line changes are made if early test flights fail to develop values reasonably close to those required by the designing engineers.

Operation – The control panel is usually arranged so that two men can observe and record all of the instrument readings at 1-min intervals. One man regulates the fuel-flow rate, while the other regulates the rate of climb. The first must be equipped with a tabulation showing variations of fuel flow with time, and the second man must have a tabulation of the altitude at which the aircraft will be at the end of each minute when climbing at rated power.

Following the heating and, as soon as the temperature of the fuel in the tank has stabilized, the vacuum pump is started with valve (12) closed and valve (19) open. The fuel pump is here presumed already to be in operation at its rated speed inasmuch as it is circulating the fuel, and now the rate of fuel flow is adjusted to the requirements of the aircraft at take-off. The fuel pressure is then set to that which will be used in the aircraft. Initial readings are taken and recorded, following which, valve (12) is manipulated after closing valve (19) in such manner as to cause the altitude reading on manometer (2f) to correspond with that of the tabulated aircraft performance at the end of each minute. If tank ram is to be maintained, valve (10) must be manipulated to maintain the assumed value of this on manometer (2c). Readings are recorded at the end of each minute.

The booster pump is usually not started until the discharge pressure of the fuel pump falls 2 psi, although some engineers prefer to allow the pressure to drop only 1 psi. As soon as the booster is started, records must be made of its discharge pressure and the input voltage must also be held to a predetermined value. In addition, some engineers require records of its current in amperes.

With the booster in operation, the climb should continue to optimum altitude at the scheduled rate without fuel-pump discharge pressure fluctuations.

The booster discharge pressure may vary slightly from time to time, but at no time should it fall to the point where it affects the discharge pressure of the fuel pump. Failure to operate to this minimum requirement indicates the need of corrective alterations.

■ Experimental Observations

Vapor Formation in Fuel Lines – The wall glass tubes were fitted in the fuel line in a simulation set-up to permit visual study of vapor formation. The bore of the glass section was made the same as that of the metal tubing to avoid changes in velocity. Fresh 100-octane fuel was then placed in the system, heated to 100 F, and an altitude run made, using the fuel pump only to simulate a simple fuel system. The progressive development of vapor in the fuel line at the fuel-pump inlet is shown in Figs. 5A to 5F. Figs. 5G and 5H show the relative vapor at two altitudes in the fuel line near the booster outlet and, therefore, immediately below and near the fuel tank. The booster is not operating in any of these views.

The test was extended beyond the failure characterized by a 1 or 2 psi fuel pressure drop to obtain the views in Figs. 5F and 5H. Both are taken at an effective altitude very near that at which the fuel boils and represent a condition that could not exist in an airplane inasmuch as the fuel pressure had dropped from 15 psi to about 4 psi. The altitude indicated in the views is uncorrected for pressure drop.

Figs. 5E and 5G show the vapor condition when a fuel pressure drop of 1 to 2 psi was observed.

Fuel and Vapor Analysis – The fuel used in this vapor formation study analyzed as follows:

	Volume, %
Butane	0.5
Iso-Pentane	17.6
Normal Pentane	4.6
Cyclo-Pentane	1.3
Iso-Hexane	2.7
Normal Hexane	6.3
Iso-Heptane	5.5
Normal Heptane and Heavier	61.5
T.E.L. – not determined	
Reid Vapor Pressure	6.7 psi

With fuel at 95 F and at an equivalent altitude of 16,000 ft, vapor samples were taken at a point near the fuel-pump inlet, which analyzed as follows:

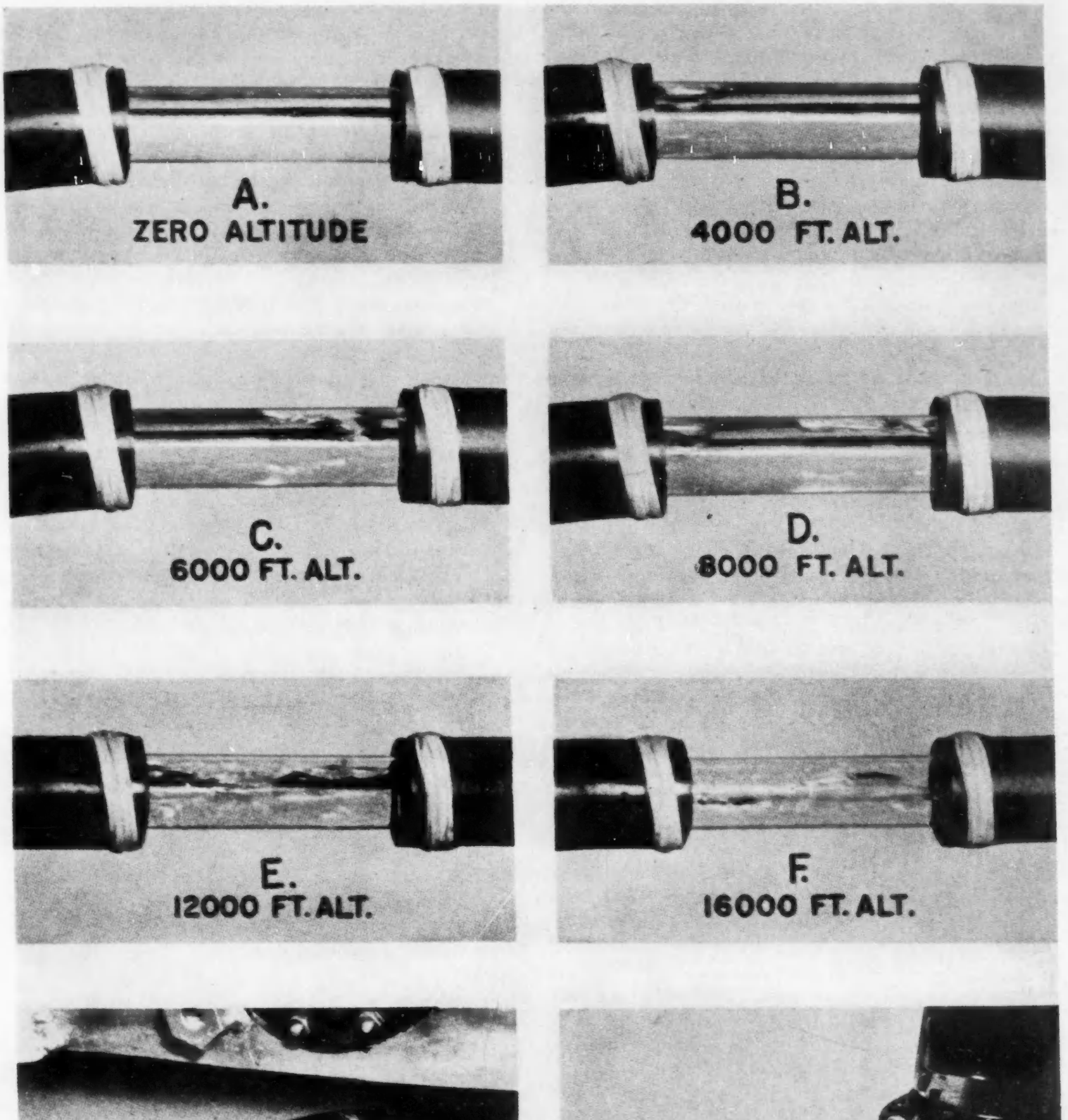
	Mole, %
Air	18.0
Butane	2.6
Iso-Pentane	62.7
Normal Pentane	12.7
Cyclo-Pentane and Heavier	4.0

Based upon the foregoing liquid composition and assuming 18 mole % air, the vapor in equilibrium with the liquid at 95 F would have the following calculated composition. The corresponding vapor pressure, including the just-mentioned quantity of air, was calculated at 382 mm absolute and was equivalent to vapor being released at an altitude of 18,000 ft:

	Mole, %
Air	18.0
Iso-Pentane	51.7
Normal Pentane	10.7
Iso-Hexane	3.4
Normal Hexane	4.3
Heptane and Heavier	7.4

II – The Centrifugal Booster Pump

About two years ago, a system of feeding fuel at high altitude was proposed which gave promise of satisfactory performance under the most adverse conditions. It consists essentially of two pumps placed in series, the first mounted on the engine and driven directly by it and the second installed directly on the fuel tank or below the tank so that



■ Fig. 5 – Progressive development of vapor in the fuel line at the fuel-pump inlet with increasing altitude (A to F, inclusive) and relative vapor at two altitudes in the fuel line near the booster outlet (G and H)

a gravity flow can be maintained to it. It has become known as the booster fuel system and has been widely adopted in this country and abroad for use on aircraft whose missions require flight at high altitude.

Many variations of the booster system have grown from the original proposal but that incorporating a centrifugal pump mounted directly on the fuel tank has consistently offered the best prospects of eliminating vapor lock in the fuel system at all altitudes and under any conditions encountered.

It was obvious early in this development that a special design of centrifugal pump was required for this duty, since the entrance of any but negligible quantity of vapor bubbles into the pump would destroy the differential pressure developed with solid liquid. Therefore, a centrifugal pump was designed with the special characteristics of ejecting the vapor bubbles formed in the fuel, and accepting only solid liquids which could then be raised to a pressure sufficient to eliminate the possibility of any further evolution of vapor in the line leading to the engine pump. A type of centrifugal pump incorporating this essential feature is pictured in Fig. 12 and its action in separating vapor from liquid is illustrated diagrammatically in Fig. 6.

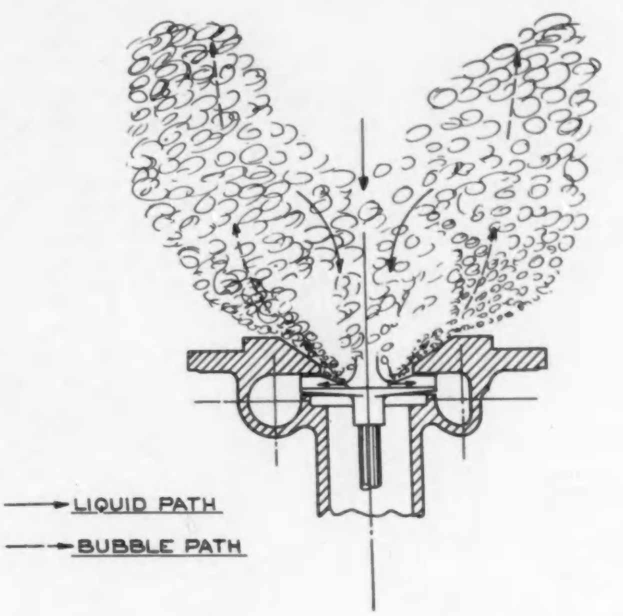
With reference to Fig. 6, a radial flow of liquid along the conical face of the throat member is induced by the inwardly extending portions of the impeller vanes. As fuel and entrained vapor bubbles flow downward toward the impeller center, bubbles are caught in this radial stream of liquid and carried away from the throat. Further bubble formation takes place when air-saturated, or superheated, liquid strikes the edges of the vanes and these bubbles, too, are trapped in the external stream and carried clear of the impeller before they can pass through into the volute. Thus, vapor bubbles approaching the throat, and those formed on contact of the liquid fuel with the impeller blades, are continuously ejected back into the body of liquid where they are free to rise to the surface and escape.

It has been recognized for some time that this first type of unit has certain performance limitations which have been difficult to determine in advance of actual installation in aircraft because of the infinite variations between fuel systems of different types of airplanes, but which may be quite definitely fixed in tests on any one fuel system. Since these limitations were known to exist and the introduction of high-performance aircraft was imminent, apparatus was set up primarily for determining comparative performance of different centrifugal pump designs, and a procedure established, after much experimentation, to use in gathering comparative performance data. The phenomena are not yet completely understood; therefore, further development of this type of booster pump has proceeded largely by the method of trial and error.

A photograph of the apparatus used in these tests appears at the top in Fig. 7A, and a diagrammatic arrangement is shown in Fig. 7 on the following page.

To show typical results produced on this test stand, three special tests were run and photographs taken of the phenomena occurring within the fuel tank under altitude conditions. The following tests were all made with aviation fuel grade 91-octane, 7 psi maximum Reid vapor pressure. The Reid vapor pressure of this fuel was actually determined to be 6.4 psi.

Photographs were taken at intervals during the three tests to record the instrument readings and show the vapor evolution phenomena under the different conditions of



■ Fig. 6 – Action of centrifugal booster pump shown in Fig. 12 in separating vapor from liquids

operation. A Kodotron light was employed flashing at approximately 2,000,000 cp for 1/35,000 sec, and the room was darkened to permit use of an open shutter for the short interval of time necessary to each shot.

■ Test I – Isolated Fuel Tank

In this test an isolated tank of fuel was subjected to reduced pressures to illustrate the vapor evolution phenomena which occur with only self-induced agitation.

Procedure – Twelve gallons of fuel were placed in the system of Fig. 7 and circulated through the heat interchanger until its temperature was raised to the desired value. With approximately 3 gal of fuel retained in the small tank *A*, this tank was disconnected from the remainder of the fuel system and subjected to reduced pressures at the desired rate.

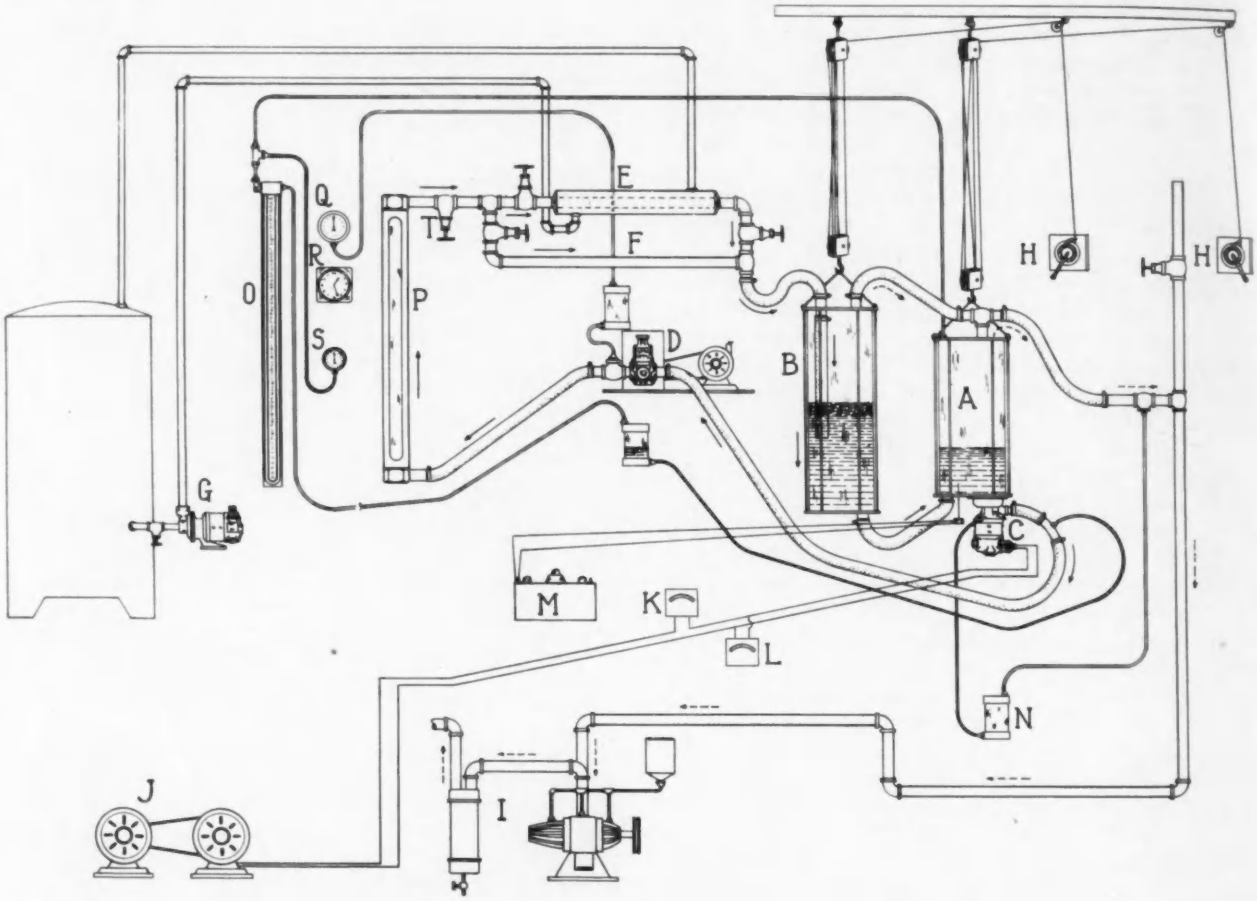
Test Conditions:

(a) Fuel
  Temperature before heating – 67 F
  Time of heating – 34 min
  Temperature at start of tests – 108.5 F
  Room temperature – 71 F

(b) Equivalent rate of climb – (to maximum altitude at which this rate could be maintained) – 4000 fpm

Results of this test are illustrated by the graph, Fig. 8, and the accompanying photographs, Figs. 8A to 8J.

The phenomena of apparent boiling due to rapid escape of air and vapor carried with it are well illustrated by the results of this test. The pressure at which air-free boiling of the fuel used occurs (at a temperature of 107 F) is reached at approximately 19,000 ft equivalent altitude. It may be observed from Fig. 8D, however, that fairly violent apparent boiling takes place at considerably lower altitudes. The altitude at which the fuel temperature curve changes most abruptly, however, corresponds well with the point at which actual boiling should start, as may be noted from Fig. 8.



■ Fig. 7 – Diagram of apparatus for determining comparative performance of different centrifugal pump designs

The reference letters in the diagram indicate the following:

A – No. 1 Fuel tank (glass cylinder).

B – No. 2 Fuel tank (glass cylinder) for adjusting level of fuel in No. 1 tank at any time during test run.

C – Centrifugal booster pump.

D – Engine pump (positive-displacement fuel pump with balanced relief valve) and independent drive.

E – Heat interchanger for heating fuel prior to test.

F – Bypass around heat interchanger used during test run.

G – Hot-water heater and pump for circulating water through heat interchanger (*F*).

H – Hoists for changing, raising and lowering fuel tanks (*A*) and (*B*).

I – Vacuum pumps and oil and fuel separators.

J – Generator – source of power for booster pump (*C*).

K – Ammeter to measure current to booster pump (*C*).

L – Voltmeter to measure EMF. impressed on booster pump (*C*).

M – Potentiometer measuring fuel temperature in tank (*A*).

N – Glass well communicating with vacuum line and booster seal chamber to prevent air from being drawn into system through seal at high equivalent altitudes.

O – Manometer measuring differential pressure across booster pump (*C*).

P – Flowmeter.

Q – Gage indicating pressure developed by engine pump (*D*).

R – Time clock to follow in adjusting rate of application of vacuum (rate of climb).

S – Altimeter indicating altitude in tank (*A*).

T – Fuel throttling valve to adjust rate of flow.

———> Fuel Flow.

------> Vapor Flow.

In this series of photographs it may be observed that a stream of bubbles is issuing almost continuously from the short nipple connected to the bottom flange of the tank. This stream is not due to air leakage into the pipe but to vapor formation on the many rough edges inside the pipe and the valve connected to it.

■ Test 2 – Booster Fuel System

This test was conducted on a simple booster fuel system identical to that shown in Fig. 7, and the standard type of centrifugal booster pump (Fig. 12), a great many of which are now in service, was used.

Procedure:

(a) The system was filled and, with the engine pump alone operating, fuel was circulated through the heat interchanger until its temperature had risen to the desired value.

(b) The engine pump delivery and discharge pressure were then adjusted to the chosen values.

(c) The vacuum pump was started and the air pressure within the tanks reduced at the desired rate.

(d) When the engine pump discharge pressure had fallen to 13 psi, because of vapor evolution in the system, the booster pump was started and the test continued to completion with this unit operating.

Test Conditions:

(a) Fuel
  Quantity used – 12 gal
  Temperature before heating – 65 F
  Time of heating – 24 min
  Temperature at start of tests – 100 F
  Room temperature – 72 F

(b) Rate of flow – 1200 lb/hr

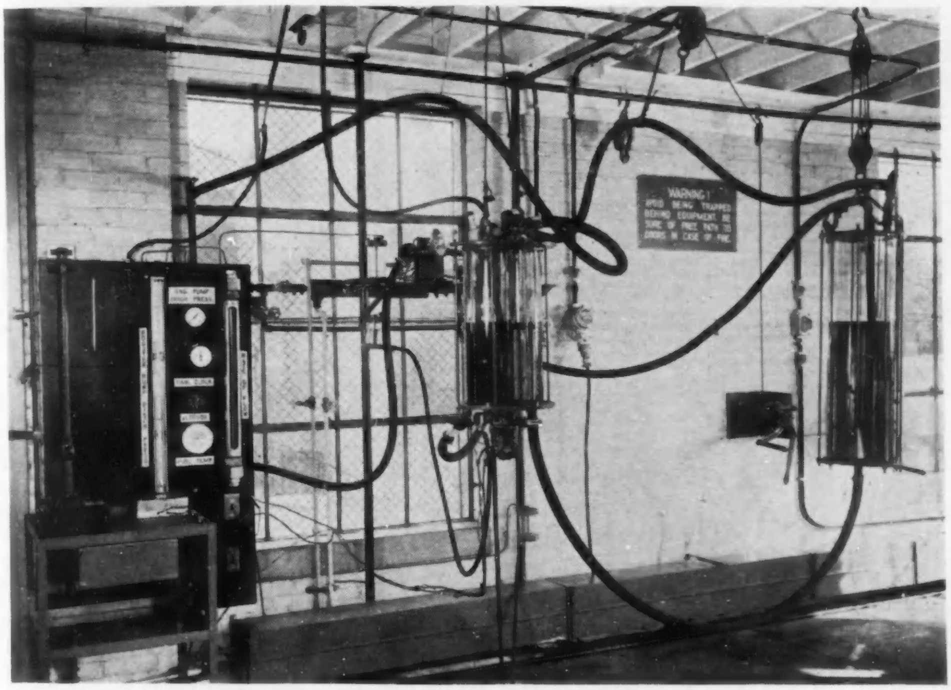
(c) Engine pump discharge pressure setting – 15 psi

(d) Equivalent rate of climb – 2000 fpm

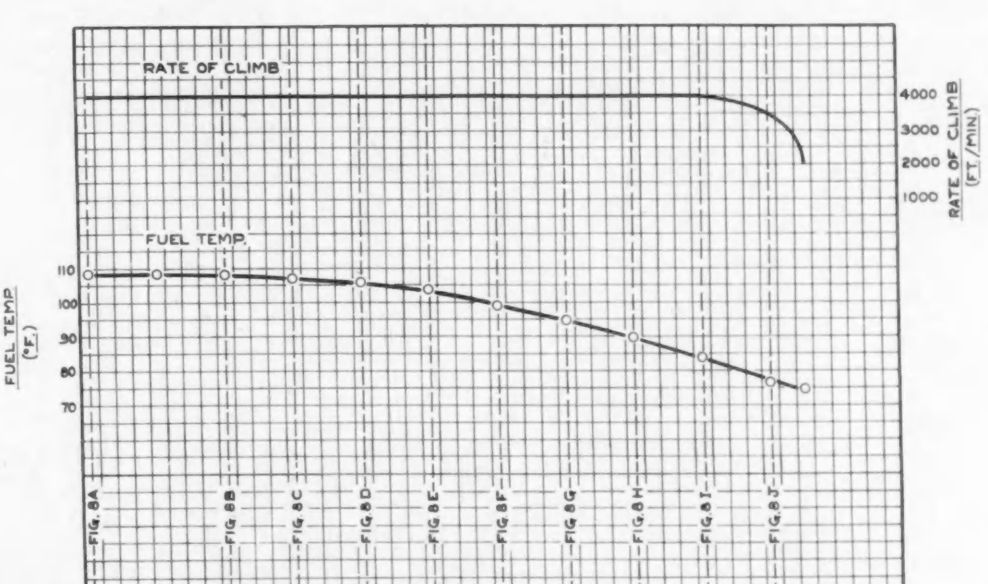
The results of this test are shown by the curves of Fig. 9 and the photographs, Figs. 9A to 9I.

Because of the influence of the booster pump impeller in agitating and assisting to stabilize the solution, and because of the reduced rate of climb and lower temperature at which the test was run, boiling of the fuel in the relatively undisturbed zone next the cylindrical walls of the tank was very little in evidence during this test.

Note from Figs. 9A and 9D, the gradual increase in number of very minute bubbles formed when the air-saturated liquid strikes the sharp edges of the impeller as it is drawn through the booster pump, by the engine pump in the early part of the test. As the alti-



■ Fig. 7A – Photograph of apparatus of Fig. 7



■ Fig. 8 – Results of test on isolated tank of fuel subjected to reduced pressure

tude is raised, an increasing quantity of these bubbles is drawn through the booster and passes into the engine pump line.

The altitude at which appreciable temperature drop starts (approximately 21,000 ft) corresponds well with that which would be predicted from consideration of the air-free boiling point of the fuel.

The growth in size of the bubbles as the altitude is increased is also well illustrated by the photographs of the test.

■ Test 3 – Booster Fuel System

This test was included to illustrate the superior performance which can be obtained with a recently developed type of fuel booster pump.

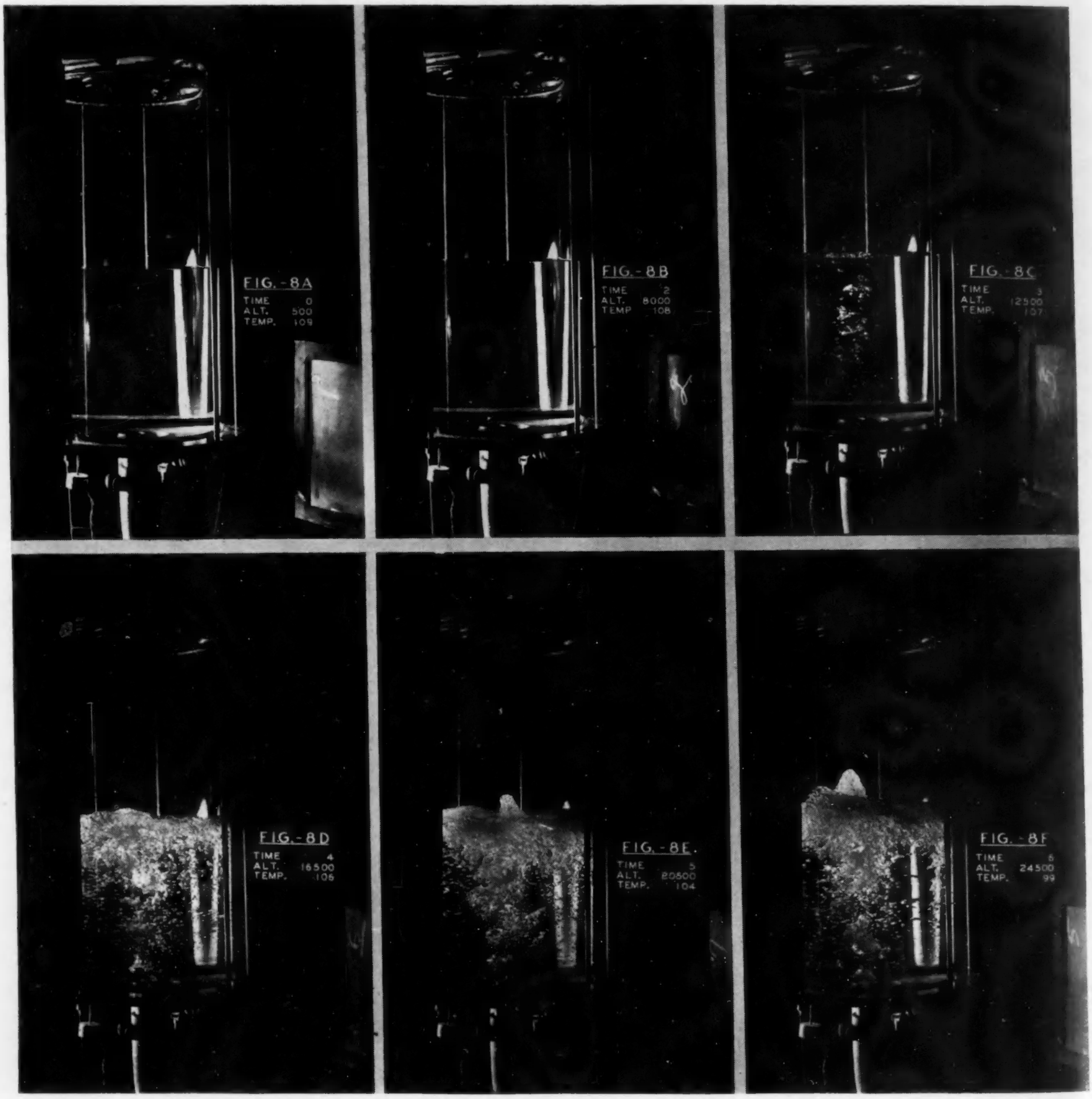
The procedure used was identical to that of Test 2, but the running conditions were much more severe. The rate of flow was increased to 1800 lb/hr, the rate of climb to 4000 fpm, and the initial fuel temperature was raised to 110 F.

Test Conditions:

(c) Fuel
  Quantity used – 12 gal
  Temperature before heating – 65 F
  Time of heating – 39 min
  Temperature at start of test – 110 F
  Room temperature – 69 F
(b) Rate of flow – 1800 lb/hr
(c) Engine pump discharge pressure setting – 15 psi
(d) Equivalent rate of climb – 4000 fpm (to maximum altitude at which this rate could be maintained)

The curves in Fig. 10 and the photographs of Figs. 10A to 10H illustrate the results of this test.

The conditions under which this test was run are be-



lieved to be more severe than any which will be encountered in flight.

Note that the formation of minute bubbles that was evident early in Test 2 does not appear in Fig. 10B, taken early in this run. Here the bubbles are forming but are being carried on through the pump with the liquid because of the higher rate of flow.

The tests just outlined are not represented to be direct altitude simulation tests of fuel systems and the data derived from them should not be considered as directly comparable to flight-test data. The influence of the following factors, which must be considered in any attempt to duplicate altitude flight conditions, has been ignored in the operation of the system used:

(a) Supersaturation of the fuel with dissolved air (resulting from relatively rapid heating).

(b) Extreme agitation (the small quantity of fuel used is continuously recirculated and agitated by the booster pump).

(c) Heat transfer from components of system. (As the fuel temperature drops at the higher altitude, due to vaporization, heat is drawn into the fuel continuously from the tanks, lines, and fittings, which have been heated to the test initial fuel temperature. Heat is also continuously supplied to the circulating fuel by the booster pump and engine pump.)

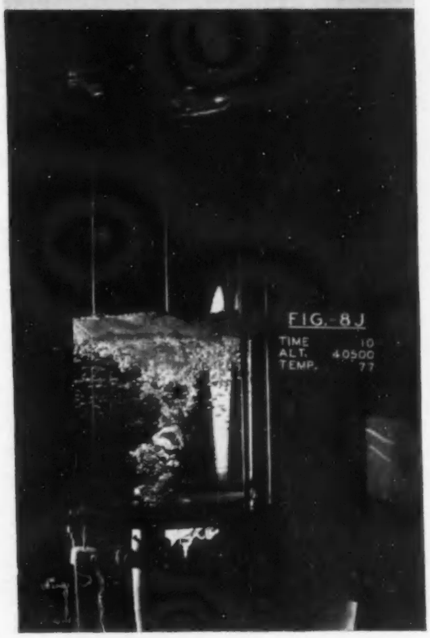
(d) Vent line effect (resulting in different rate of pressure change within the tank than indicated by equivalent rate of climb).

To bring out more clearly the action of the booster pump of Fig. 6, a series of photographs was taken with the conditions of delivery rate and altitude varied to develop the phenomena desired.

Fig. 11A was taken with the booster pump impeller acting on a slightly air supersaturated fuel at very low altitude pressure and shows the pattern of minute bubbles developed by the open blades of the impeller.

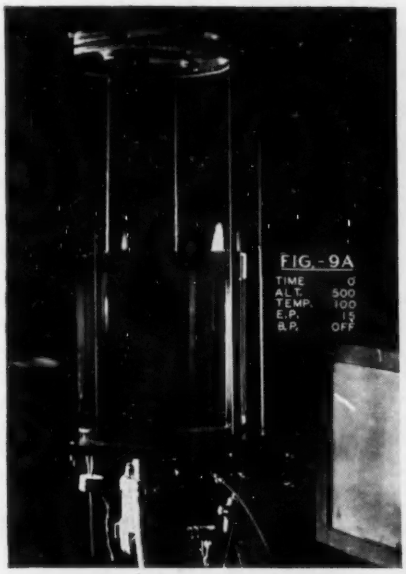
Although these pictures were taken with an exposure time of 1/35,000 sec, this was not fast enough to stop completely the impeller and bubbles trapped in the high velocity stream leaving it.

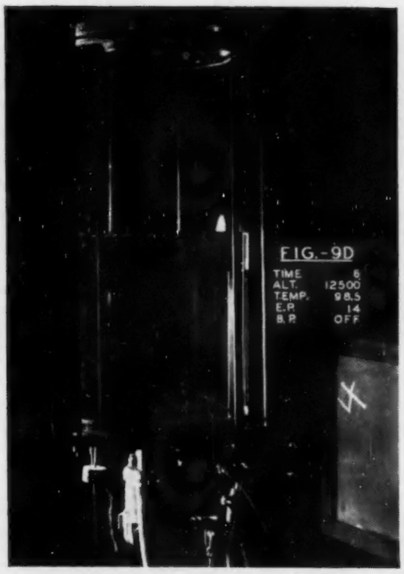
The photographs, Figs. 11B to 11E inclusive, were taken under conditions such that a progressive approach to failure is pictured. This was accomplished in this particular in-

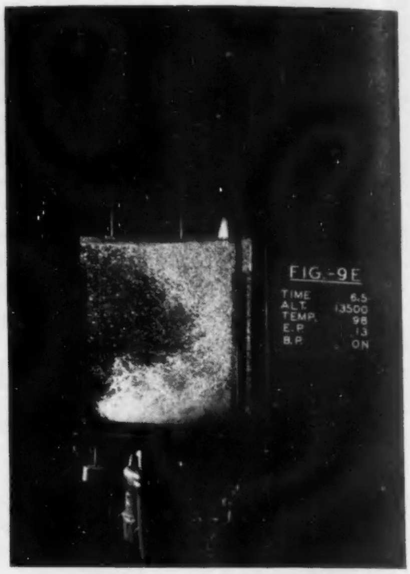


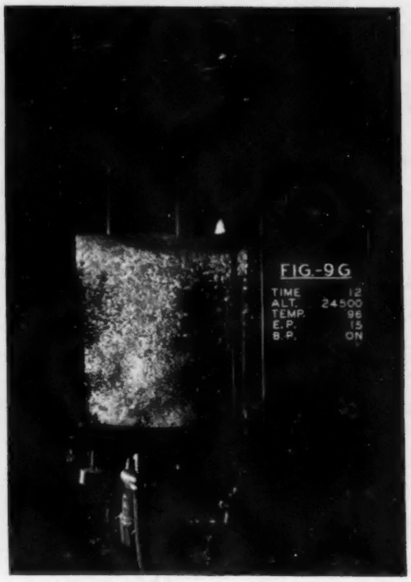
Figs. 8A to 8J inclusive – Photographs of vapor-evolution phenomena and instrument readings in test on isolated tank of fuel subjected to reduced pressure

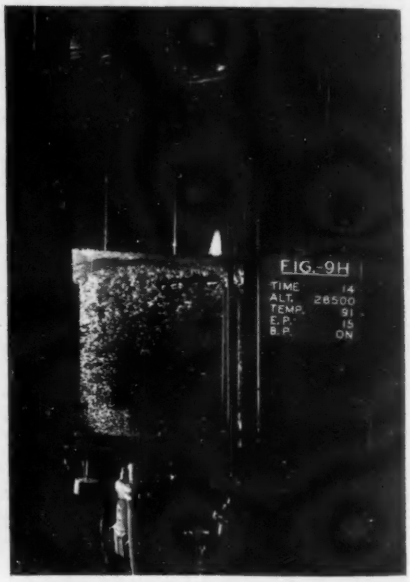
Photograph caption explanation:
Time – Min from start of test
Alt.– Altitude above sea level, ft
Temp. – Temperature of the fuel, F





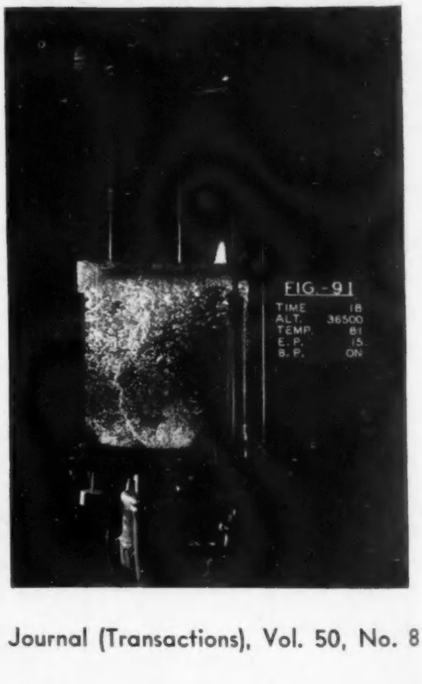






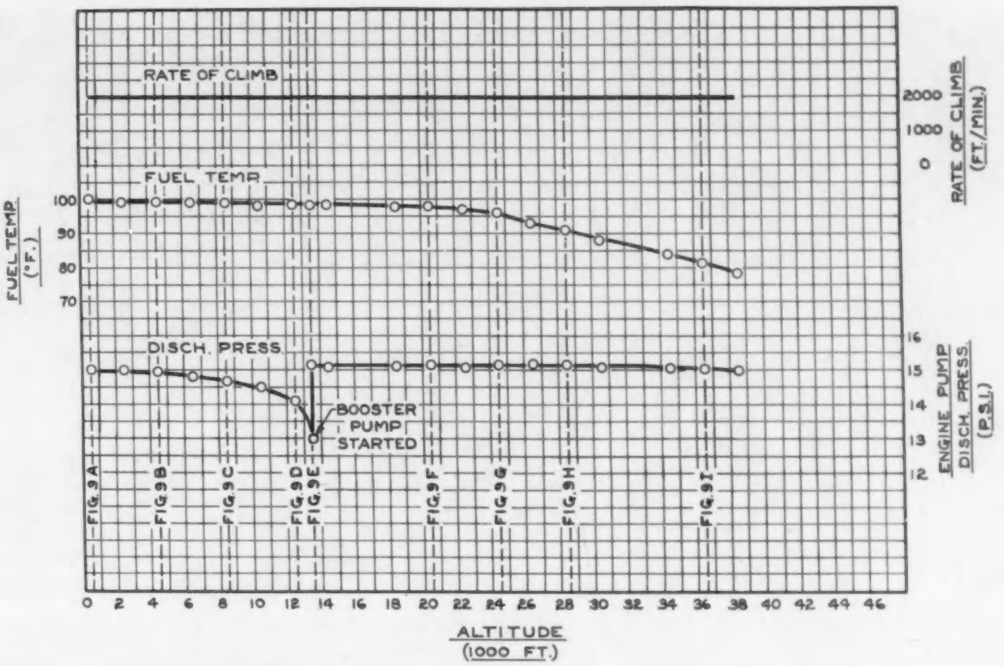
■ **Figs. 9A, 9D to 9I inclusive – Photographs of vapor-evolution phenomena and instrument readings in Test 2 of booster fuel system**

Photograph caption explanation:
Time – Min or min and sec from start of test
Alt. – Altitude above sea level, ft
Temp. – Temperature of the fuel, F
E.P. – Discharge pressure of engine pump, psi
B.P. – "OFF," booster pump not operating; "ON," operating



stance by gradually increasing the delivery rate while holding the tank pressure constant. The identical phenomena may occur whenever the vapor-separating capacity of the unit is exceeded, whether this is due to the characteristics of the fuel, fuel temperature, flow rate, rate of climb, or any combination of these factors or others influencing the altitude performance.

●

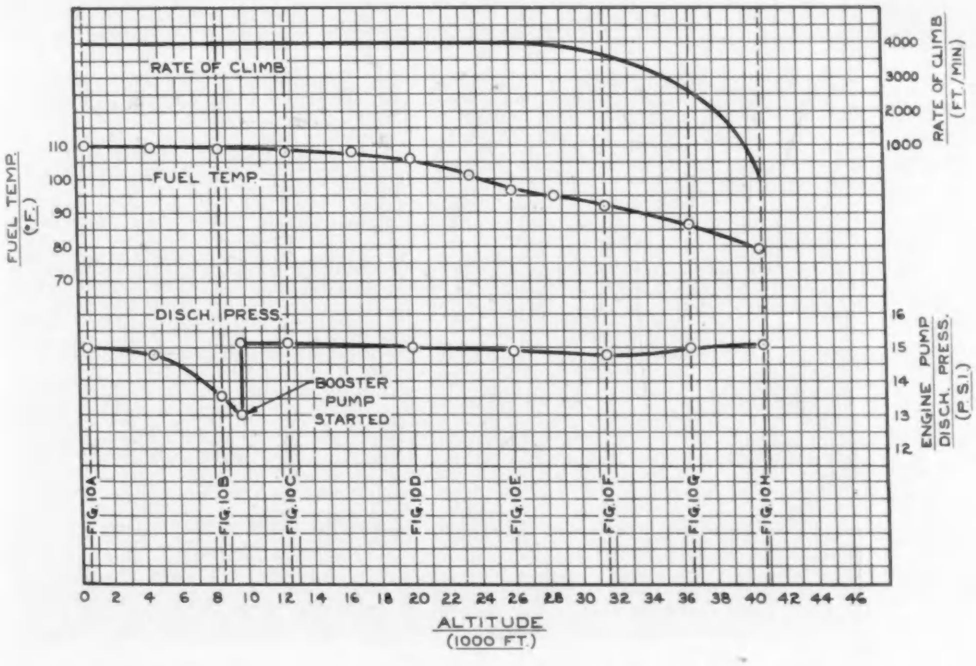


■ Fig. 9 – Results of Test 2 on booster fuel system

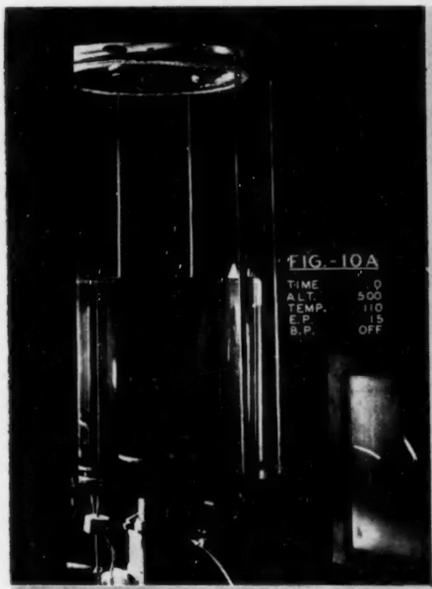
In order to obtain pictures which would not be obscured by the bubble formation within the tank, independent of the impeller, exposures were made each time, just prior to the point at which vapor would be evolved in quantity from the relatively quiescent liquid near the walls of the tank.

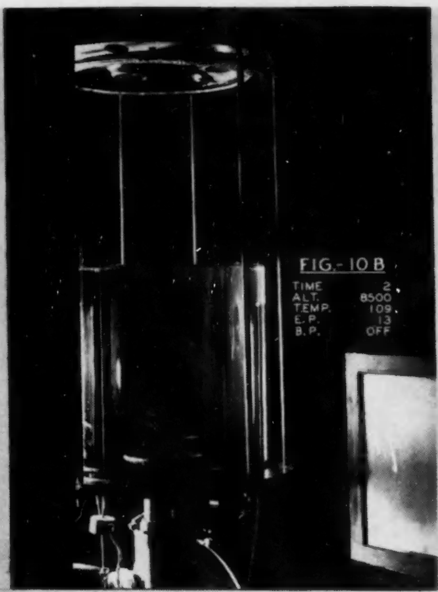
The normal action of the unit before the bubble pattern has been influenced by approaching failure is illustrated by the photograph, Fig. 11B. The minute size of the bubbles formed at the impeller, and their progressive growth as they leave the impeller and as their velocity lessens, should be noted. As the delivery rate is increased, the radial stream of liquid carrying the bubbles flows at reduced velocity until a point is finally reached at which this external stream disappears. When this occurs, the impeller is enveloped in a blanket of vapor (see Fig. 11E) and it ceases entirely to pump. Also, at this same instant, it ceases to cause any disturbance whatever to the liquid in the tank.

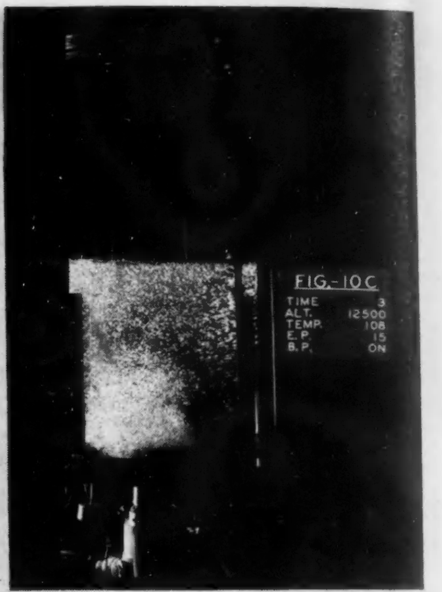
Acknowledgment, with sincere thanks, is made for the cooperation of Messrs. H. C. Britten, chief technologist, and W. A. A. Beaver, office technologist, both of Shell Oil Co., Inc., Wilmington, Calif.

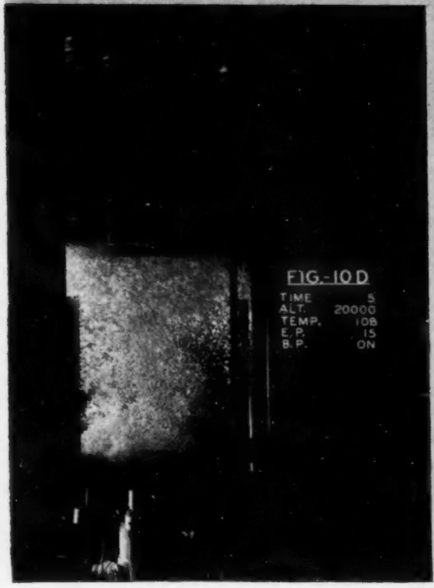


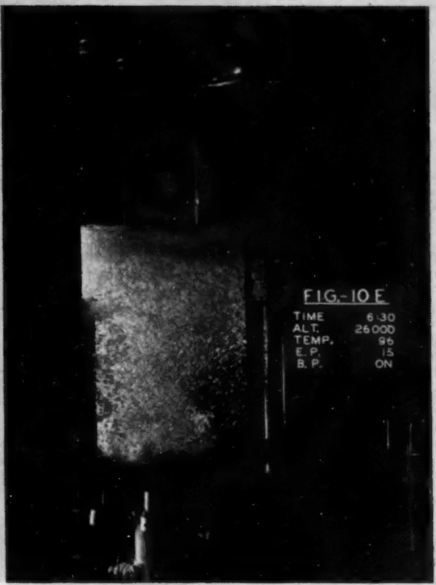
■ Fig. 10 – Results of Test 3 on booster fuel system





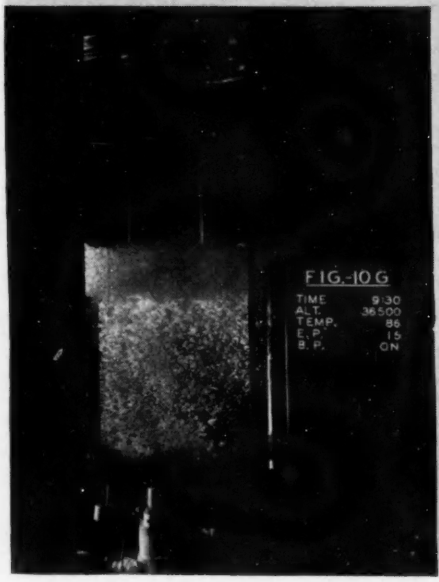


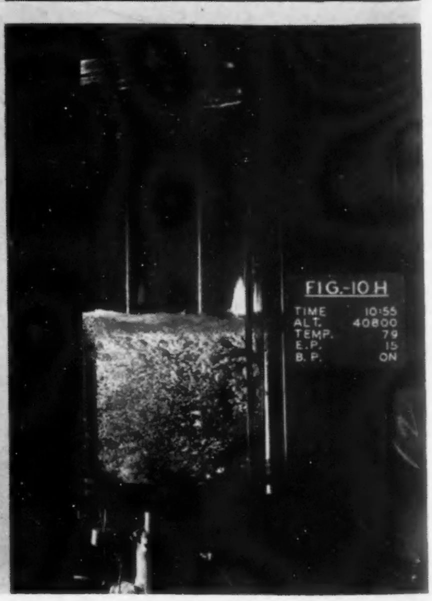




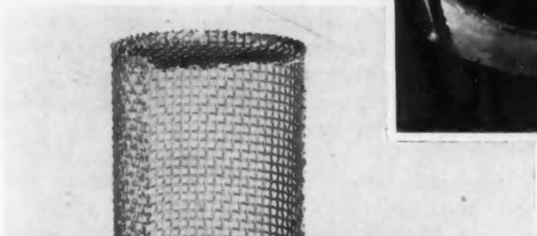
■ Figs. 10A to 10H inclusive – Photographs of vapor-evolution phenomena and instrument readings in Test 3 of booster fuel system

Photograph caption explanation:
- Time – Min or min and sec from start of test
- Alt. – Altitude above sea level, ft
- Temp. – Temperature of the fuel, F
- E.P. – Discharge pressure of engine pump, psi
- B.P. – "OFF," booster pump not operating; "ON," operating





■ Figs. 11A to 11E inclusive – Series of photographs taken to illustrate action of booster pump shown in Fig. 6, with conditions of delivery rate and altitude varied to develop the phenomena desired



■ Fig. 12 – Centrifugal booster pump which ejects vapor bubbles formed in fuel and accepts only solid liquids

List of References

(1) "Test Procedure and General Information in Current Use in the Development and Utilization of Aviation, Motor, and Automotive Diesel Fuels," compiled by Cooperative Fuel Research Committee, May, 1941, Edition.

(2) "Handbook of Petroleum, Asphalt, and Natural Gas," Kansas City Testing Laboratory, Bulletin No. 25, 1931 Revision.

(3) "Investigation of Vapor Lock in Aviation Fuel Systems," CFR Progress Reports, O. C. Bridgeman, SAE Journal, November, 1938, p. 20; March, 1939, p. 26; November, 1939, p. 21.

(4) "The Effect of Airplane Fuel Line Design on Vapor Lock," by O. C. Bridgeman and H. S. White, SAE Journal, October, 1930, pp. 444-450, 458.

(5) "Airplane Fuel Line Temperature," by O. C. Bridgeman, C. A. Ross, and H. S. White, SAE Journal, August, 1931, pp. 121-125.

(6) "The Effect of Weathering on the Vapor-Locking Tendencies of Gasoline," by O. C. Bridgeman and E. W. Aldrich, SAE Journal, September, 1930, pp. 344-351.

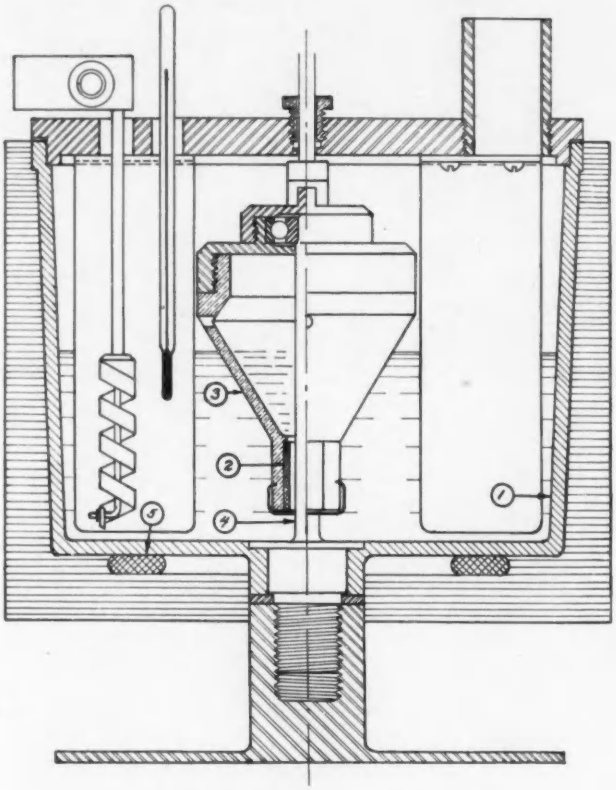
(7) "The Vapor-Locking Tendencies of Aviation Gasolines," by O. C. Bridgeman and H. S. White, SAE Journal, August, 1930, pp. 218-233.

(8) "The Properties of Gasolines with Reference to Vapor Lock," by O. C. Bridgeman and E. W. Aldrich, SAE Journal, July, 1930, pp. 93-110.

(9) "Vapor Pressures of Gas-Free Gasolines," API Bulletin, Jan. 2, 1930.

AN OIL

(A Simplified Apparatus for



■ Fig. 1 – Original corrosion tester

THE stability of engine oils at high temperature, as indicated by their tendency to corrode bearing metals, can be tested in the laboratory, and results which correlate satisfactorily with engine tests can be obtained, according to these authors. But they warn that the apparatus does not eliminate the necessity for engine tests, inasmuch as there may be other factors tending to affect the correlation.

The apparatus for testing is said to save time and expense. A number of samples can be rated quickly under a variety of conditions; corrodibility of bearing metals or the corrosiveness of oils can be predicted approximately, thus reducing the number of engine tests needed. Likewise, oils can be "weeded" to leave only those worth engine testing. Uniformity of oils or bearings can be tested to enable check of subsequent samples with the initial, tested samples, while the effect of changing engine operation variables may be simulated so that the stability of oils can be indicated under these operating conditions.

Taking oils whose behavior was known in aircraft engines in test-house operation, Pratt & Whitney Aircraft has made tests with such satisfactory correlation in the relative order of their corrosiveness and tendency to sludge, that they now use the apparatus to distinguish between good and bad oils for aircraft engines in test-house service, the authors report.

The development of the apparatus and methods of test used are thoroughly covered in this paper.

■ ■ ■

THE AUTHORS: NEIL MacCOULL (M '17), during the past 23 years with The Texas Co., has carried out research investigations on the performance and application of motor fuels and lubricants, some of which have been presented in papers before the SAE. An active member of the Society, he was chairman of the Metropolitan Section in 1925-26; vice president representing Fuels and Lubricants in 1940; and has been a member of the SAE Research Committee since 1927. EARLE A. RYDER (M '11), consulting engineer for Pratt & Whitney Aircraft, has worked on aircraft-engine designing and testing since 1914. His early experience was with Simplex Automobile Co. and Aeromarine Plane Co. Joining Pratt & Whitney in 1925, when it was organized, he has been with that company ever since on design, testing, and research. A. C. SCHOLP, after graduation from Stevens Institute of Technology in 1936 with an M.E. degree, attended MIT, obtaining his Ph.D. in Physical Chemistry in 1939. Since that time he has been connected with The Texas Co. in the engineering research department of the Beacon Laboratory.

IN the development of automotive and aircraft engines much effort has been put on the establishing of simplified bench tests to minimize the full-scale engine testing that is so time-consuming and expensive. The final answer in any situation must be obtained from the engine itself but, in preliminary work, time can often be saved by testing one detail at a time. In the same way, the improvement of oils having suitable characteristics for a given type of operation can be hastened by the use of short-time bench tests directed to the examination of certain specific properties. The apparatus to be described has been under investigation over a period of years for the purpose of testing, at high temperatures, the stability of engine oils as indicated by their tendency to corrode bearing metals.

In starting the present development, the following features were considered desirable and incorporated in the design:

1. The few parts in contact with the test oil should be easily cleanable.

2. A simple pumping device should be provided which would spray the heated test oil through an air space in finely divided form.

3. The metal specimen, whose weight loss would indicate the amount of corrosion, should be easily removable for weighing.

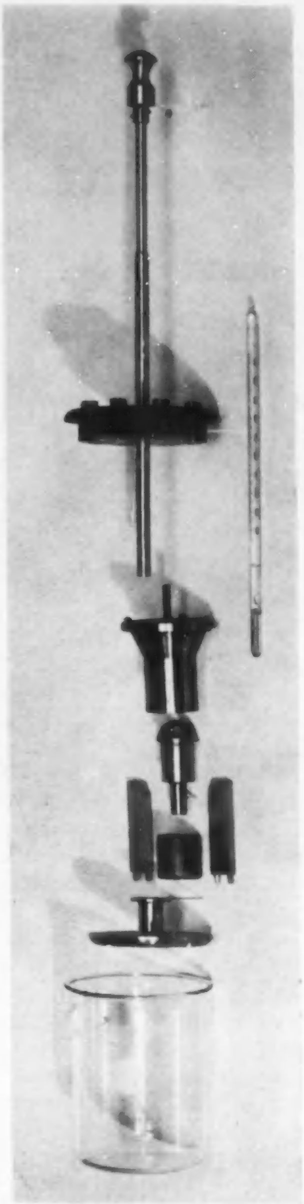
4. Bearings should be made of the same stock used for commercial engines so as to obtain the uniformity of large-scale manufacture.

In 1935, an apparatus embodying the foregoing features

CORROSION TESTER

Testing Corrodibility of Bearings and Corrosiveness of Lubricants)

by NEIL MacCOULL,* E. A. RYDER,†
and A. C. SCHOLP*



■ Fig. 2 – Essential parts of corrosion tester as used by Pratt & Whitney Aircraft

was built in The Texas Co. laboratories. Fig. 1 shows the original unit, which consisted of a spinner (3), partially submerged in the test oil, driving the test bearing (2), which rotated about the spindle (4). The spinner was centered at the top by a ball bearing and was driven by an electric motor through a dog-clutch. Oil was prevented from swirling with the spinner by the baffles shown. Heat was applied by the chromalox ring (5), controlled by a conventional laboratory thermostat. In subsequent units the ball bearing was discarded because it accumulated oxidation products from the oils and was difficult to clean. In operation, capillarity and centrifugal force raised the oil and sprayed it from the holes in the periphery of the spinner through the atmosphere in the upper part of the test pot. The thorough and fine division of the test oil, which was thus accomplished, provided a large area of contact between the oil and air and was an important feature of this design. Further light has recently been thrown on this subject by other experiments.[1]

Oil, replacing that which was discharged, caused a circulation between the bearing (2) carried by the spinner and the stationary spindle. The high rate of oil shear in the bearing assured an intimate scrubbing, which promoted corrosion if the oil developed any corrosive material.

Experience with this apparatus indicated the need for drastic cleaning between runs. It was necessary to abrade the iron pot surface with emery cloth or scour with hydrochloric acid in order to control catalysis, and to remove some of the oil "dopes" which were tested, so that they would not carry over and influence subsequent runs. The usual cleansing with solvents was not sufficient. This demonstrated the desirability of using glass containers.

Recently in connection with a study of the relative corrosiveness of various bearing materials, Pratt & Whitney Aircraft recognized that the MacCoull apparatus was well adapted for this purpose and undertook to design a similar machine, retaining the original features and adding the following:

1. Provision for multiple tests (10 at a time).
2. Use of a smaller sample of oil.
3. Use of glass beakers.
4. Provision for solid catalysts.
5. Ratio of oil volume to bearing area similar to that obtained in full-scale engines.

Such an apparatus was designed and several have been built. Operations have been conducted independently by Pratt & Whitney Aircraft and The Texas Co. with good agreement as to qualitative results, although the set-up of the test parts is slightly different in the two companies.

In the apparatus designed by Pratt & Whitney Aircraft, the *outside* of the bearing was used as the test surface, this being more convenient in working with bearing metals which can be electroplated, such as most of the pure metals and some of the alloys. Fig. 2 shows the working parts of the apparatus as used by Pratt & Whitney Aircraft.

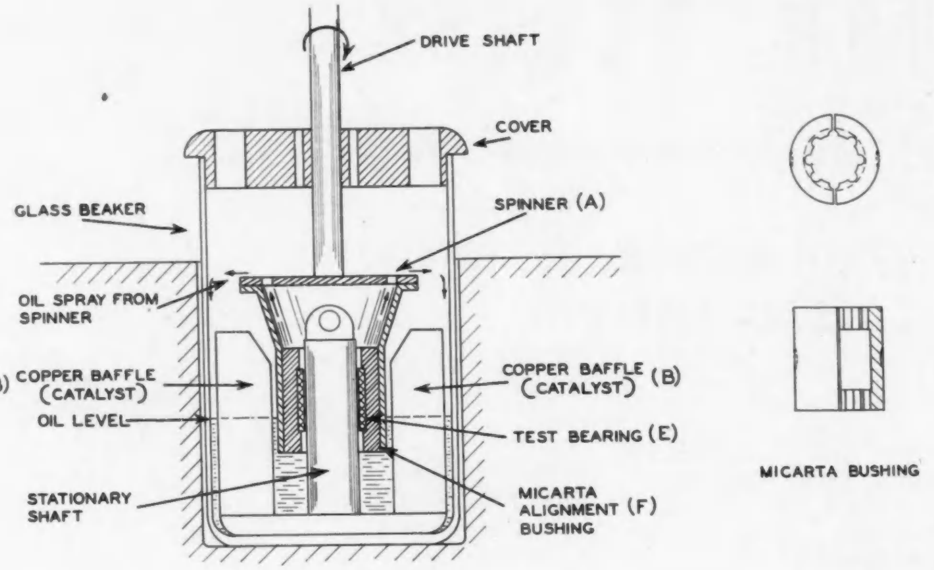
The test bushing is 1 in., outside diameter, by 1 in. long and is made with the proper running clearance in the spinner. The bearing is mounted on the stationary pin with a very slight taper so that, when the parts are removed from the beaker and washed, tapping the bearing support on the bench releases the bearing which then can be weighed. Much work has been done on the relative corrodibility of metals and, for judging the suitability of oils for production test-stand use, a lead surface has been adopted. This surface is easily produced by electro-plating, but precautions must be taken to assure that all specimens are plated by exactly the same technique. Otherwise, differences in the corrosion rate will be noticed.

In The Texas Co.'s version of this apparatus the same type of bearings is used as in the original unit of Fig. 1. This has been considered an important feature in an apparatus planned specifically for tests on oils rather than on bearing

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 16, 1942.]

* The Texas Co.
† Pratt & Whitney Aircraft, Division of United Aircraft Corp.

[1] See *Industrial and Engineering Chemistry*, Analytical Edition, Vol. 13, pp. 317-321: "Indiana Stirring Oxidation Test for Lubricating Oils," by G. G. Lamb, C. M. Loane, and J. W. Gaynor.



■ Fig. 3 – Assembly of one element of corrosion tester as used by The Texas Co.

metals because of the necessity of using a large supply of bearings which are similar to commercial alloy bearings, are uniform, and reasonable in cost. Figs. 3 and 4 show the details of a unit using this type of bearing. It will be seen that a split micarta bushing *F* holds the bearing *E* in the stainless-steel spinner *A*. These bushings are provided with internal flanges which center the bearing on the spindle and prevent contact with it. These flanges are serrated so as not to impede the oil circulation. It has been found that preventing actual contact between the bearing and the spindle has been an important feature in obtaining reproducible data. When this feature was not used, wear of the bearing was added to the corrosion loss and the wear varied with slight differences in alignment.

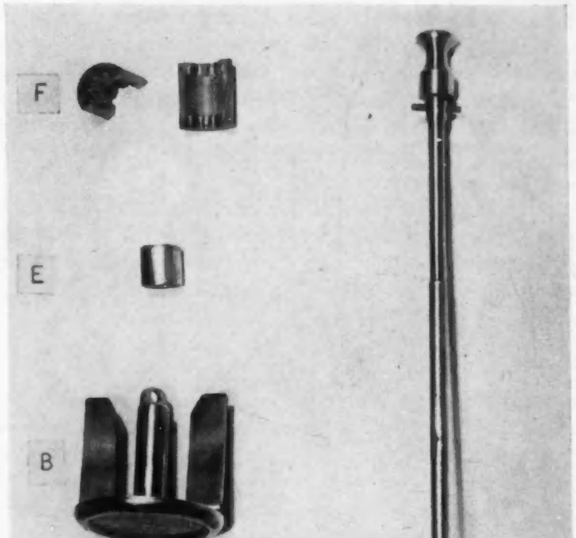
Except for these differences, most of the remaining features of the apparatus as used by The Texas Co. and by Pratt & Whitney Aircraft are identical.

Figs. 5 and 6 show the complete apparatus with ten corrosion units, which may be run simultaneously.

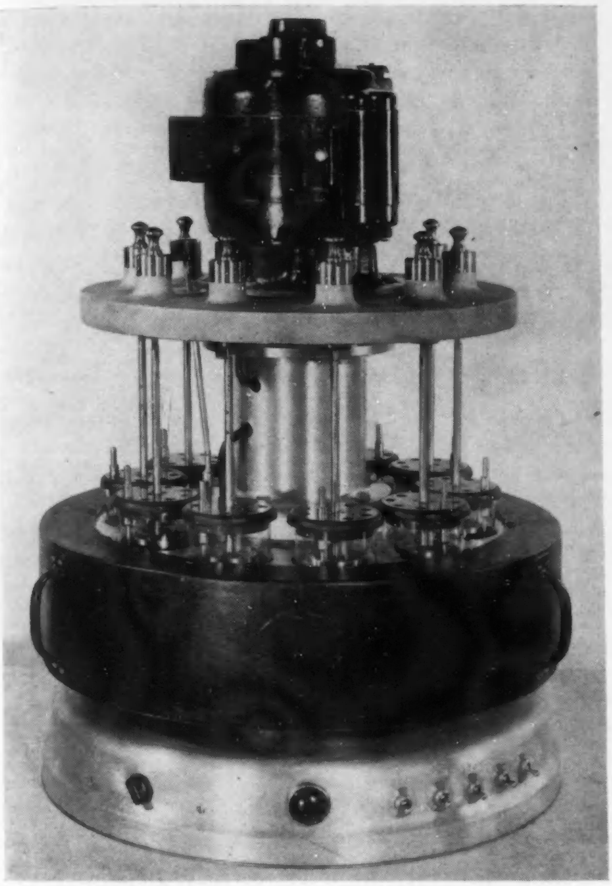
The test oils are placed in glass beakers set into an electrically heated aluminum block which can be maintained at the desired temperature. A single motor at the top of the apparatus drives ten spindles, each of which is attached to one of the spinners. Any of these spindles may be raised by hand at any time, disconnecting it from the drive pulley, and allowing access to the parts in the bottom of the beaker.

Motion of the bearing draws oil through the bearing clearance, whence it travels up the conical side of the spinner and is discharged from holes in the top of the spinner near the periphery, as previously mentioned. The rate of pumping does not seem to be critical but, since the spindles all operate at the same speed and the bearing clearance is held to the same value on all test pieces, fairly uniform pumping action is obtained.

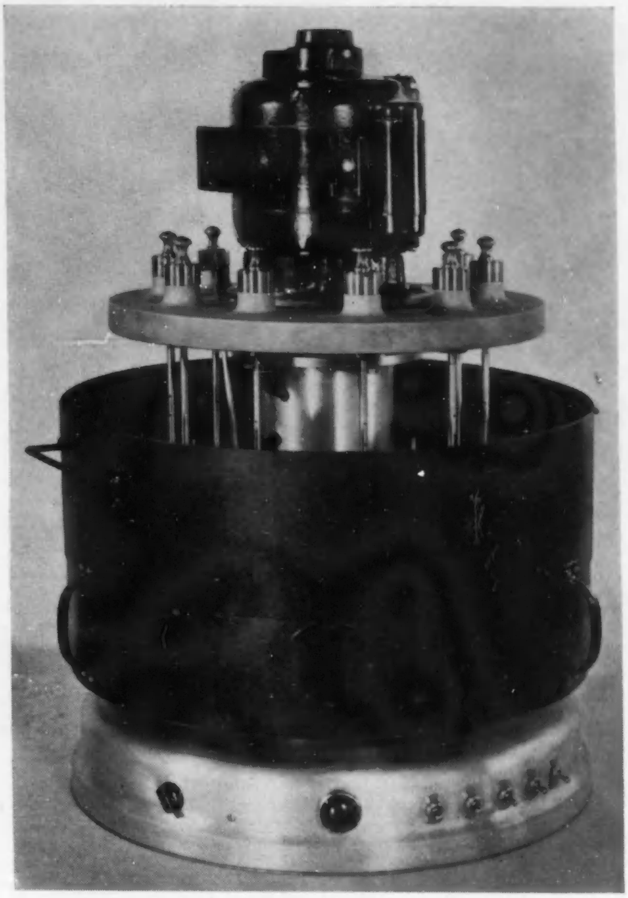
Air circulation through the beaker is produced also by the rotation of the spinner, which draws air in through holes near the center of the cover and discharges it through holes near the periphery. The number and size of the holes, which control the air circulation, have been selected to give an air-flow rate such that reasonable variations do not markedly alter the results. Too low a supply of air



■ Fig. 4 – Essential parts of corrosion-tester assembly shown in Fig. 3



■ Fig. 5 – Complete MacCoull corrosion tester with 10 test beakers – guard ring removed



■ Fig. 6 – Complete MacCoul corrosion tester as operated

retards the oil oxidation and thus the corrosion. Too great a supply also reduces the corrosion, possibly because it carries away some of the volatile and more corrosive of the oxidation products.

The two baffles (*B*, Figs. 3 and 4) prevent excessive rotation of the mass of oil in the beaker. Since all other metal in contact with the test oil except the bearing is stainless steel, catalytic action may be concentrated at these surfaces. These can be made of any material, but at present copper is used and the baffles are polished immediately before each run.

■ Test Procedure

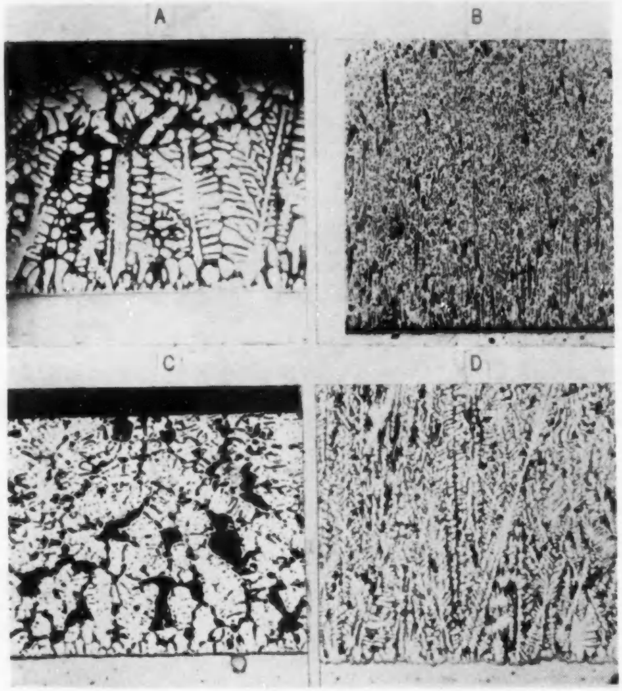
In setting up for a test, 125 cc of the oil to be tested are placed in each beaker, this being sufficient to cover the test bearing. To facilitate heat transfer, the space around each beaker is filled with a heavy oil. The spinners are lowered into the beakers, the covers put in place, and, when the proper temperature is obtained, the motor is started, revolving all ten spinners at 3000 rpm. It is considered good practice to obtain duplicate runs for each test oil, and thus at least two beakers are usually filled with each test oil. The bearing assembly is rotated for 2 hr, after which the bearing is removed, cleaned by scrubbing in Underwood solution[2], and weighed, the corrosion weight loss being measured as the difference between this weight and the weight of the original unused bearing.

This cleaning operation is very important since the oxidation of some oils produces resinous substances which deposit on the bearing surface to such an extent as to protect it from further corrosion. This has been demonstrated where 10-hr runs have been made without intervening cleaning periods, and weight loss was considerably less than on corresponding runs where the bearings were cleaned each 2 hr. Further weighings are made at 4, 6, 8, and 10 hr, and the cumulative weight loss of the bearings for each of these periods is recorded.

■ Test Bearings

The bearings used by The Texas Co. in the experiments which follow, are ⅝ in. inside diameter by ¾ in. long, and are made from the same copper-lead-coated steel strip which is used for the manufacture of commercial automotive engine bearings. After blanks for bearings are stamped from this strip, they are formed to shape and finally broached or diamond-bored to diameter. Continuous improvement in commercial copper-lead bearings has reduced their sensitivity to corrosion. For a time, the bearing manufacturer supplied the laboratory special bearings which corroded as rapidly as some early commercial copper-lead bearings. These were made by a change in the quenching rate of the copper-lead alloy as it was poured on the continuous strip of steel during the regular production

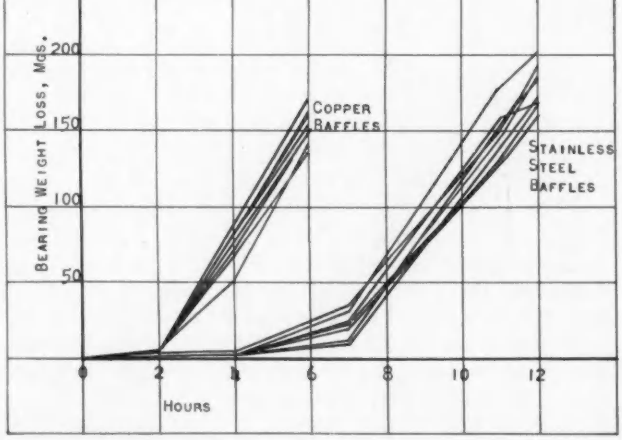
[2] Mixture of equal volumes of methyl acetate, toluol, and denatured alcohol. Other solvent solutions as, for example, 50% methyl ethyl ketone and 50% 86-deg naphtha, may be used.



■ **Fig. 7 – Photomicrographs of typical copper-lead bearings (reduced from photomicrographs taken at 100X)**

A, early, very corrodible commercial bearing; **B**, modern commercial bearing, far less corrodible; **C**, special bearing made for the corrosion tester, of an alloy similar in composition to **B** but with a coarser microstructure which makes it more corrodible; **D**, aircraft bearing manufactured by an entirely different process from the other bearings shown

of commercial bearings. The metallurgical differences in bearing structure which are involved are indicated in Fig. 7 which shows: (*A*) an early commercial bearing which corroded rapidly both in extreme engine service and in this test apparatus; (*B*) a typical modern bearing which is much less affected by corrosion; and (*C*) a similar alloy to (*B*) but with a different quenching rate that resulted in test bearings with a coarse microstructure which accelerates the corrosion phenomena. Present practice is to use the *B* structure for tests in this apparatus because bearings made from it are more uniform in their tendency to corrode



■ **Fig. 8 – Corrosion-time curves – comparison of baffle materials at 350 F**

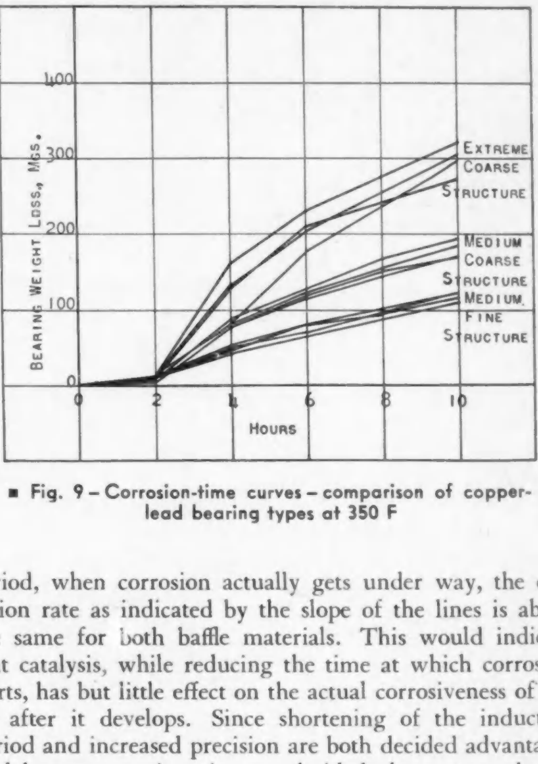
than the *C* structure, although the latter are known to have a higher rate of corrosion under given conditions. As a matter of interest, the structure of a typical copper-lead aircraft bearing is shown in *D* for comparison.

■ Reproducibility of Results

Fig. 8 indicates the reproducibility which has been obtained by this test. In testing a large number of samples of the same oil, it has been found that corrosion weight losses can generally be reproduced to within 10% of the average value of a number of runs.

■ Effect of Catalytic Surfaces

The baffles *B* of Figs. 3 and 4 may be of various materials. Runs with polished copper as well as stainless steel, which cover a rather wide range of catalytic effect, also are shown in Fig. 8. It will be seen that the copper baffles have not only reduced the induction period from 6 to 7 hr, to 2 hr, but that the termination of the induction period is indicated more precisely. Note that, after the induction



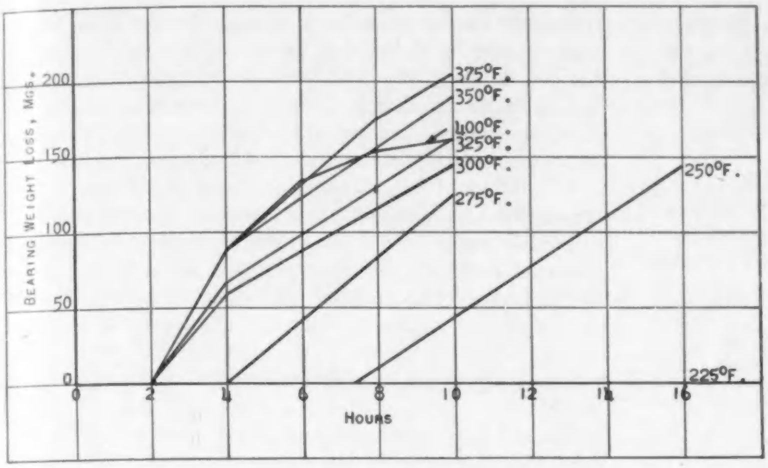
■ **Fig. 9 – Corrosion-time curves – comparison of copper-lead bearing types at 350 F**

period, when corrosion actually gets under way, the corrosion rate as indicated by the slope of the lines is about the same for both baffle materials. This would indicate that catalysis, while reducing the time at which corrosion starts, has but little effect on the actual corrosiveness of the oil after it develops. Since shortening of the induction period and increased precision are both decided advantages in laboratory testing, it was decided that copper baffles should be used as standard equipment.

It may be seen that this apparatus might be used also for testing the relative effectiveness of various metals as catalysts for oil oxidation simply by comparing lengths of induction periods.

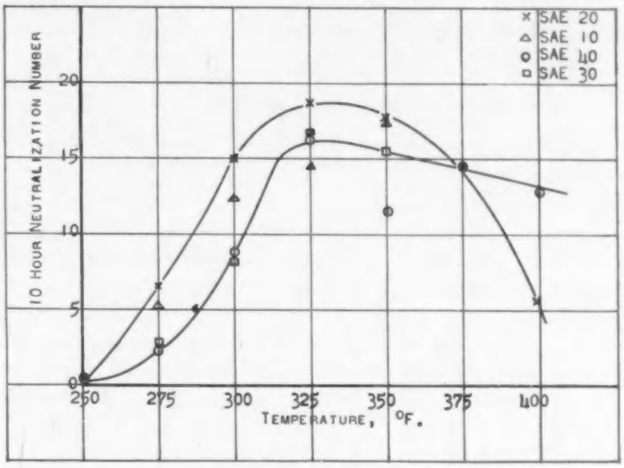
■ Effect of Copper-Lead Structure

Fig. 9 shows the effect on corrosion of changes in the microstructure of copper-lead bearings. These structures were shown in Fig. 7 and cover a considerable range in the size and distribution of lead particles in the copper matrix. It will be seen that, the coarser the structure, the greater the weight losses. This result would naturally be expected



■ Fig. 10 (left) – Corrosion-time curves at various temperatures – coarse-structure copper-lead bearings – copper baffles

■ Fig. 11 (below) – Neutralization number versus temperature

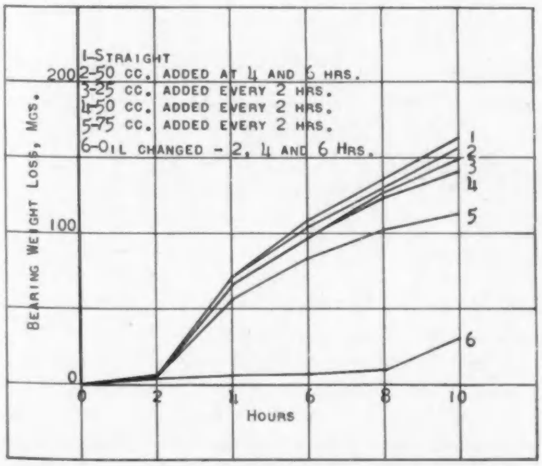


since the coarser structures have larger agglomerates of lead at the bearing surface available for attack by the oil. Note that, with these widely different microstructures, there was but little change in induction period.

■ Effect of Temperature

Fig. 10 shows the effect of temperature on a series of tests run on an SAE 20 oil. At the lowest temperature (225 F) no appreciable bearing weight loss was obtained during the test period while, at higher temperatures, the induction period became shorter and the corrosion rate immediately after the induction period became progressively greater. At 375 F, and more notably at 400 F, a change in curve shape indicating a different phenomenon began to take effect. While the initial corrosion rate was as high as at any other temperature, this rate fell off rapidly so that after 6 hr the total corrosion was less than at 350 F. This result may have been due to several causes, such as:

1. The rapid increase in viscosity as the oil oxidized at this temperature, thereby slowing the circulation rate;
2. Deposition of material on the bearing which protected it from corrosion; or
3. Volatile acids being driven off at the higher temperatures.



■ Fig. 12 – Corrosion-time curves – coarse structure copper-lead bearings at 350 F – copper baffles

The corresponding changes in final neutralization number over this temperature range are shown in Fig. 11. It may be seen that the effect of increasing temperature up to about 350 F was to increase the neutralization number but, above this temperature, a decrease was noted. This is similar to the effect of temperature on corrosion.

■ Effect of Oil Additions

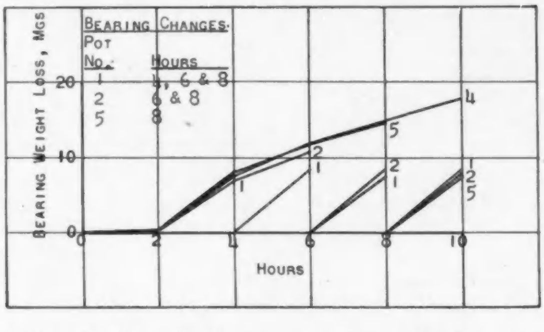
Experimental runs were made in which various quantities of oil were withdrawn periodically during the corrosion tests and replaced by fresh oil. This simulated engine operation where fresh oil is added periodically to make up for the normal oil consumption of the engine. Results are shown in Fig. 12. Only a small reduction in corrosion was effected unless more than 40% (50 cc) of the volume under test, were replaced each 2 hr. Even replacing 60% (75 cc) every 2 hr caused only a moderate reduction in corrosion. This indicates that variations in quantity of periodic fresh oil additions to test engines, caused by normal variations in oil consumption rate, probably have but little influence on the rate of bearing corrosion, and this conclusion seems to be in agreement with experience.

The only marked reduction in corrosion caused by fresh oil additions occurred when all the oil was changed at the end of each induction period. The bottom curve of Fig. 12 shows the effect of such a procedure, oil being changed

completely at 2, 4, and 6 hr. The oil was not changed at 8 hr, and it will be seen that the oil change at 6 hr, after the normal induction period of 2 hr, started corrosion much as usual.

Effect of Bearing Changes

Runs were made in which used bearings were replaced by new bearings but without changing the oil, according to the schedule indicated in Fig. 13. In Pot No. 1 new bearings were installed at 4, 6, and 8 hr. The corrosion of each of these bearings over the subsequent 2-hr period was nearly the same as for the original bearing during the first 2 hr after the induction period. This would indicate that:



■ Fig. 13 – Corrosion-time curves – coarse structure copper-lead bearings – copper baffles

1. The decreasing rate of corrosion normally experienced after the 4-hr period is probably due to a reduction in lead available for further corrosion at the bearing surface.

2. The corrosiveness of the oil after the induction period does not increase with time, but remains constant, at least until the end of these 10-hr runs.

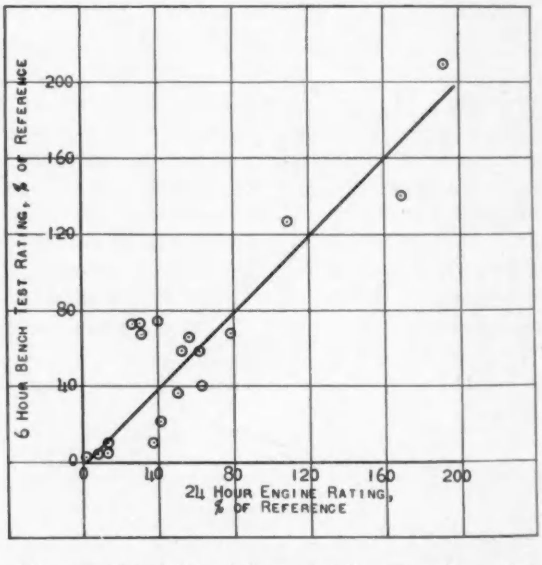
Correlation with Full-Scale Engines

The chief value of any small-scale corrosion test is its ability to predict the corrosion behavior of an oil in the commercial engine in which it is subsequently used. It has been found that the corrosion test described here gives corrosion results which agree approximately with automotive-engine corrosion tests, under laboratory operation with similar oil temperature within the limits of reproducibility of the engine tests. A high-temperature engine corrosion test was conducted in a laboratory on a commercial multicylinder engine with connecting-rod bearings made from the same copper-lead strip used in fabricating the corrosion test bearings and used in most of the tests reported here with this apparatus. The crankcase oil temperature was held at 300 F for a period of 24 hr by means of electrical heaters. A reference oil was used between runs on test oils, both for rating purposes and to check engine conditions. Ratings were expressed on a percentage basis, by comparing the bearing weight loss from a run with the test oil, to the average weight loss from bracketing runs with a reference oil. Thus, a rating of 60% would mean that the weight loss due to corrosion from the test oil was 60% of the average weight loss of the two runs on the reference oil immediately preceding and following the test oil. Corresponding ratings were made on the corrosion apparatus with the same test oil and reference oil after 6 hr at 350 F. This time was found to give the best correlation, and the temperature was probably a close approximation to the temperature of oil leaving the connecting-rod bearings in the engine test with a crankcase oil temperature of 300 F.

Correlations with oils, including those containing additives, under this procedure are shown in Fig. 14. It may be seen for the particular data shown that, in no case, are there substantial deviations from the 45-deg line of perfect correlation. Since the accuracy of the engine data was considered to be $\pm$ 20% at best, the agreement between the results from this apparatus and the corresponding engine tests seems quite satisfactory. We wish to emphasize, however, that this apparatus does not eliminate the necessity for engine tests, inasmuch as there may be other factors which will have a tendency to affect the corrosion correlation.

The test apparatus, however, does furnish a research tool for weeding out oils so that engine testing may be reduced to a minimum.

Tests have been run on this apparatus by Pratt & Whitney Aircraft on oils whose behavior was known in aircraft engines in test-house operation. Such satisfactory correlation has been obtained in rating these oils in the relative order of their corrosiveness and tendency to sludge, that they now use it as a test to distinguish between good and bad oils for aircraft engines in test-house service. Oils are rated on a basis of corrosion, viscosity increase, and rise in neutralization number after a run in this apparatus.



■ Fig. 14 – Correlation between engine corrosion rating and bench-test corrosion rating –

$$\text{Rating} = 100 \frac{\text{Test Oil Weight Loss}}{\text{Average Reference Oil Weight Loss}}$$

Usefulness of Test

The corrosion test described here has several features of special interest. These may be enumerated as follows:

1. A large number of samples may be rated quickly for corrosion behavior under a variety of conditions.

2. By using proper test conditions, the corrodibility of bearing metals, or the corrosiveness of oils may be approximately predicted under high-temperature operation of commercial engines so that the total number of engine tests may be appreciably reduced. Conversely, a large num-

ber of oils or bearing materials may be "weeded" so that engine tests are made only on those materials which indicate promise from a corrosion standpoint.

3. The test may be used to check the *uniformity* of different samples of oils or bearings and insure that subsequent samples conform in corrosion characteristics to results obtained on initial samples.

4. The effect of changing engine operating variables such as rate and magnitude of oil additions, oil temperature variations, variation of bearings, and so on, may be simulated in this test, so that the stability of oils may be indicated under these operating conditions.

Future Developments

It is fully realized that further development of this apparatus is desirable. The authors believe that, with modifications, this apparatus would be suitable for considerably greater uses than the tests reported in this paper. For instance, it may be developed into a much-needed test for the tendency of oil deterioration products to deposit "varnish," such as may be found on pistons under certain conditions of engine operation. Tests run with means for subjecting the bearings to considerable load would indicate if "varnish" or protective films might be deposited on bearings which would be tenacious enough to remain under severe engine operation. In addition, analyses of the used oils from runs in this apparatus for neutralization number, naphtha insolubles, carbon residue, and viscosity increase, may possibly be correlated with tests on crankcase oils from standard multicylinder engine runs. Studies of oil behavior in atmospheres other than air may be very profitable, especially those representing exhaust or blowby gases which are produced by various fuels under various conditions of combustion.

Conclusions

Even with all the future prospects of expanding the usefulness of this apparatus, it appears that accomplishments so far have resulted in the development of a reproducible corrosion test which evaluates oils of a wide variety of base stocks and "additives," in approximately the order in which they tend to prevent corrosion of alloy bearings in automotive-type engines under high-temperature laboratory tests, and aircraft engines in test-house operation.

The apparatus described here has, in addition, the advantage of short tests, can be quickly and thoroughly cleaned, and may be used for a wide variety of temperature conditions. Since ten units may be operated simultaneously, a large number of tests may be obtained simultaneously under similar conditions.

Acknowledgment

The authors wish to acknowledge the splendid cooperation furnished by The Cleveland Graphite Bronze Co. in supplying various types of test bearings, and the personal interest and suggestions given by John K. Anthony of that concern.

Crashproof Fuel Tanks

THE principles involved in crashproof fuel tank construction can be fairly well established. The tank should be able, if possible, to absorb the forces transmitted to it due to the impact, and the fluid pressure waves thereby set up, without rupture. The material or combination of materials comprising the tank shell should obviously combine good elastic properties with high resistance to impact and, at the same time, be electrically non-conductive and gasoline resistant. The tank should also have some degree of self-sealing qualities, be as light as possible, easily formed, and not subject to intensive crack formation upon impact. Baffles should be provided, except in the case of very small tanks, to break up and dissipate the formation of pressure waves and should be of non-conductive material. The tank flanges and connections for fittings would require special attention. These should preferably consist of an integral portion of the tank shell suitably shaped to the desired contour. Metal inserts might be used if built into the material of the shell and suitably protected thereby.

A very important factor in the ability of the tank to resist rupture will, of course, be the manner in which it is mounted and installed.

A means of support that would insure low unit loading on the tank and provide some degree of flexibility, such as a padded cradle or wide padded straps, is highly desirable. Precautions would have to be taken to prevent overflowing fuel from spilling on the tanks or its supports as such a condition could nullify all efforts at protection. All feed, vent, drain and other lines attaching to the tank would naturally be of suitable non-metallic flexible hose and all connections to the tank would have to be given due consideration as to the possible effects they might introduce to induce rupture of the tank. Attention would also have to be given to proper ventilation of the space surrounding the tank so as to preclude the formation of dangerous gasoline-air mixtures in the tank compartment.

Modern bullet-proof and self-sealing tanks comply of course with the foregoing principles, their general construction comprising a thin inner liner of gasoline resistant synthetic rubber surrounded by a thick layer of sealant, such as pure gum rubber, encased by a tough elastic outside cover of vulcanized rubber or leather which acts as a compression member for the sealant. Another outside cover, perhaps of compressed fiber or metal, may be added to give additional strength. However, these tanks are heavy, bulky, and costly, and their application to civil aircraft would seriously affect existing operational economy and efficiency by reducing payload and range.

It is therefore believed that the modern bullet proof and self-sealing type of fuel tank does not offer the best solution to the problem of providing adequate protection to civil aircraft against explosion and fire in a crash and that a better means to secure the desired end would be to press the development of a crashproof tank which would be more applicable to the needs of such aircraft.

Excerpts from the paper of the same title by J. W. Baird, Civil Aeronautics Administration, presented at the National Aeronautic Meeting of the Society, New York, N. Y., March 12, 1942.

PROGRESS in Structural STRAIN-GAGE

THE technique of the application of modern electrical strain-gage equipment to basic problems in modern aircraft structural design is discussed in this paper. The general nature of the equipment is very briefly described. Special arrangements of strain-gage units for each type of stress are discussed together with problems of economy and efficiency in the use of the available equipment.

The application of these methods to a number of typical problems is outlined to point out means by which more precise design can be accomplished. The knowledge, so gained, permits the simplification of the aircraft structure with a corresponding simplification of production problems. These methods are explained from the point of view of their application to the structural behavior of complete structural assemblies.

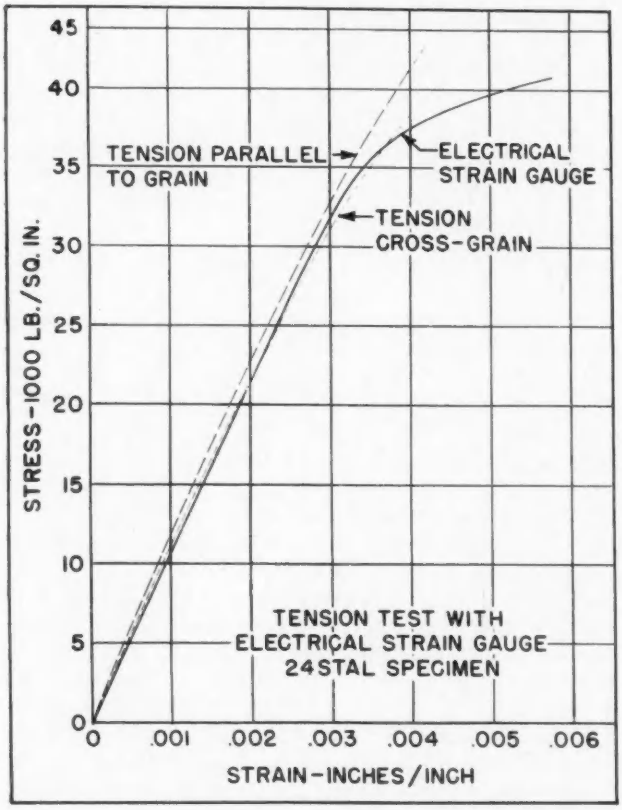
Most of the discussion is concerned with static proof tests or flight tests, but some comment on laboratory applications in connection with the development of structural theory is made. Finally, a summary of the difficulties involved and the problems that may be anticipated terminates the paper.

* * *

THE AUTHOR: C. R. STRANG, has been strength engineer with Douglas Aircraft Co., Inc. since 1936. Prior to that he had been employed in the engineering departments of Chance-Vought Corp., Consolidated Aircraft Corp., and Hall-Aluminum Aircraft Corp. From 1931-1933, Mr. Strang was a consultant to the Bureau of Aeronautics, Navy Department, Washington, D.C. He was graduated from the University of Michigan with a B.S. in Aeronautical Engineering in 1928.

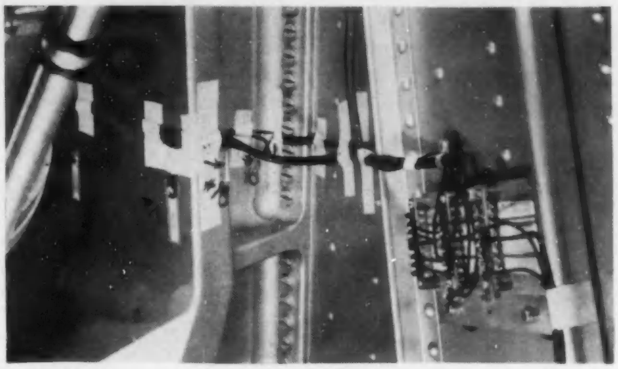
STRAIN-GAGE technique might, at first glance, seem so highly technical a subject as to occasion some surprise at interest in it at a time when production problems seem to require all of our attention. Actually, of course, the greatest possible aid to increased ease in production is to start with the original design and simplify the structure itself, resulting naturally in a simplification of the entire production problem right from the outset. Better understanding of the problems involved already has allowed considerable progress in the matter of reducing the complexity of aircraft structures. So far, every effort has been made to limit the application of the techniques about to be discussed to those problems for which solutions are needed most urgently, always with the idea in mind of reducing the factors of ignorance involved to the point that we are permitted to simplify the structures which may be in question. The very words "strain gage" infer a primary concern with theory, due largely to their past limitation to laboratory work. They are, however, here discussed as a work-a-day aid to the structural designer, substantiating past decisions and supplying information for better future decisions. In general, these methods, in a halting sort of way, provide a seventh sense that permits the designer to see and understand the inner workings of a structure under load.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 12, 1942.]



■ Fig. 1 – Comparison of stress-strain curve indicated by electrical strain-gage equipment with curves determined by more customary means

Design through TECHNIQUE



■ Fig. 2 – Part of arrangement of strain gages used to determine the stress distribution across the unsymmetrical I-section of a large forging

by C. R. STRANG
Douglas Aircraft Co., Inc.

■ Equipment

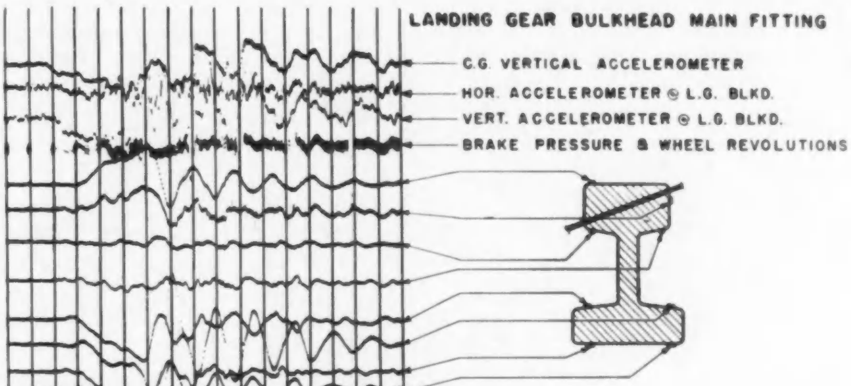
I will limit myself almost entirely to the application of electrical resistance strain-gage apparatus to the solution of typical structural design problems in order to make clear the vastly accelerated progress made possible by the use of this equipment. The resistance units themselves have been thoroughly described by deForest and Leaderman in NACA Technical Note No. 744, and a detailed description of the associated amplifying and recording apparatus as developed by the Douglas Aircraft Co., was presented by Messrs. Root, Dickinson and Ostergren in a paper which was presented before the Institute of Aeronautical Sciences, Jan. 27, 1942.

In general, the equipment consists of resistance units of fine wire cemented to the stressed surface under investigation and connected in a suitable bridge circuit. The cross-section, and hence the resistance of the gage unit, is varied by the elongations or contractions to which it is subjected. The resulting unbalance of the bridge circuit is amplified by electronic means, and recorded by equipment suitable to the nature of the tests; a recording oscillograph is used if the tests are dynamic in nature, or stylus recorders, or even simple visual meter readings are used if the stresses involved are static.

Much of the substantiation of existing structural theory has been due to the use of mechanical strain gages of one sort or another. For the most part, such gages unfortunately need to be used with considerable understanding as they require a highly developed technique of their own. Their nature is such that good results with them are usually only obtained with highly trained personnel and under very favorable laboratory conditions. In actual aircraft work, the odd sections to which gages must be attached, the lack of room for their installation, the rapid variation of stresses in flight or dynamic test work, and the importance of stresses in thin sheet working in the wave state, to which suitable attachment of a mechanical gage is very difficult, have all resulted in very limited use of such gages. By contrast, the electrical gage units are themselves very small and suffer from none of the foregoing disadvantages. In addition, when they are used with suitable associated apparatus, they are remote reading and self-recording. For each type of stress, such as simple tension or compression, or bending stresses, or stresses due to shear, there is a special most-suitable use of the gage units. These special applications for each type of stress will first be considered.

■ Axial Stresses

In all work with these gages, it is necessary to remember that a strain and not a stress is being measured. Therefore, Poisson's ratio corrections must be made wherever



■ Fig. 3 – Record of stress distribution across I-section of landing-gear bulkhead main fitting taken during a landing

■

stresses not parallel to the axis of the gage exist. Where simple axial loads with a uniform stress distribution across a cross-section of a member could be assumed to exist, a single gage would be adequate to determine that stress. Such a fortunate state of affairs is difficult to achieve in the laboratory, let alone in actual aircraft structures and, for that reason, a much more satisfactory procedure is to use a number of gages so interconnected that an average of the stresses imposed on all of the gages is read. By this method, a true indication of the average axial stress and of the true axial load in the member may be obtained.

Assuming purely axial stresses parallel to the axis of the gages, the stress will of course be

$$\text{Stress} = E \times \text{Strain}$$

Fig. 1 is a comparison of a stress-strain curve indicated by electrical gage equipment with curves determined by more customary means. Note that, at stresses above 37,000 psi, the electrical gages indicate a much larger increase in strain than should be expected. The useful range of the gages is, therefore, taken to be of the order of 0.0037 in./in. strain. Whence:

Gage limit for dural = 10,000,000 x 0.0037 = 37,000 psi
Gage limit for steel = 29,000,000 x 0.0037 = 107,000 psi.

These figures indicate that the gages are suitable for proof tests and flight-test work in which yield stresses are not exceeded for 24 STAL dural or equivalent, and for steel up to ultimate tensile strengths of the order of 125,000 psi.

For flight-test work or static proof tests on aluminum alloys, the indicated range is satisfactory. The permissible upper limit is unfortunately not high enough to permit direct simple use beyond the yield stresses in destruction test work.

For investigations on the more highly heat-treated steels, it is sometimes possible to widen the apparent useful range by taking advantage of preload stresses that may exist. For instance, in a typical investigation of a landing-gear axle for a large airplane, for which the ultimate design load factor might be 4, the axle could be steel, heat-treated to 180,000 psi and worked at the ultimate load factor to an allowable modulus of rupture of the order to 250,000 psi. In that case, the airplane simply parked on the ground fully loaded would be experiencing axle stresses as high as 250,000/4 = 62,500 psi. A gage installed at such a point while so stressed would not be overstressed when the load was removed as in flight. In such a case, if a landing were made at the full expected applied load factor or $\frac{4}{1.5} =$ 2.667, the increment of stress imposed on the gage would be $\frac{2.667 - 1}{4} \times 250{,}000 = 104{,}000$ psi. This would still be within the permissible working range of the gage. The preceding outline is merely an example, the actual case would not be quite so simple, but it serves to illustrate the fact that the gage unit's range of usefulness can sometimes be extended by special circumstances.

* See: (1) NACA Technical Note 709, 1939: "A Semi-Graphical Method for Analyzing Strains Measured on Three or Four Gage Lines Intersecting at 45 Deg." by H. N. Hill; (2) Douglas S. M. 3126: "Strain Star Formulas," by W. B. Klemperer; (3) *Journal of the Aeronautical Sciences*, Vol. 7, July, 1940, pp. 403-404: "The Tensor Gage," by W. B. Klemperer; (4) U. S. Department of Commerce, Bureau of Standards Research Paper No. 559: "The Determination of Stresses from Strains on Three Intersecting Gage Lines;" (5) *Journal of the Aeronautical Sciences*, Vol. 7, August, 1940, pp. 438-40: "Circles of Strain," by J. A. Wise.

Bending Stresses

Where a varying stress distribution is known to exist, and it is desired to determine that distribution, a number of individual gages located at critical points may be used. From these data, a plot of the stress distribution across the section may be made, and bending and direct stresses may be isolated subsequently.

One such case is illustrated by the arrangement used to determine the stress distribution across the unsymmetrical I-section of an extremely large, forged fitting, part of which is shown by Fig. 2. Pickups were distributed as shown, together with a section of the record taken during a landing, by Fig. 3. For convenience, these circuits are usually so arranged that tensile stresses are indicated by an upward swing of the trace and *vice versa* for compression. The vertical lines are time lines, 0.10 sec time interval being indicated by each space. The redistribution of bending stress due to beam curvature is another field that has been investigated by a similar installation.

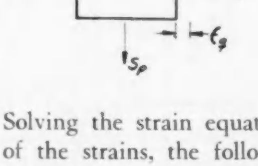
When bending stresses are the sole concern, it is possible to use two gages on opposite sides of a bending member, interconnected in a single circuit so that the total stress differential (a combination of the tension on one side, plus the compression on the other side) is indicated. Such applications are particularly accurate because the nature of the circuit is such that the two gages are used on opposite legs of the bridge. No extra dummy gages are necessary, and automatically perfect temperature compensation is obtained. Wherever the nature of the problem is such that the determination of a total bending differential stress is satisfactory for the purpose, such an installation is ideal.

Shear Stresses

In the general case, with the direction and magnitude of the principal stresses unknown, it is always possible to determine the complete state of stress on a surface if a sufficient number of strains directionally disposed at sufficiently large angles to each other are known.

Literature covering the mathematics of the determination of the state of stress from such data is fairly complete, having been developed originally for use with mechanical gages.* Of all the available methods, the use of three gages, each forming an angle of 60 deg with the other, in the shape of a delta, as shown by Fig. 4, lends itself most easily to rapid mathematical reduction. The result sought is the determination of the direction and magnitude of the principal stresses from which any desired components, such as the shearing stresses, may be established. A sketchy development of the mathematics involved is as follows:

An element oriented in such a manner that only principal stresses exist is considered. The strain in each direction follows:



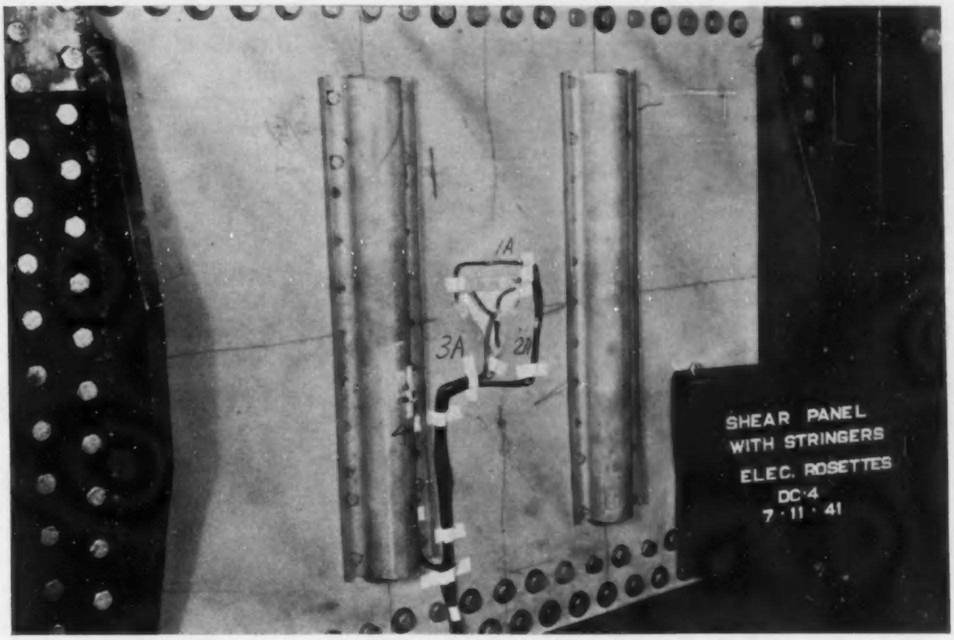
$$\epsilon_p = \frac{S_p}{E} - \mu \frac{S_q}{E}$$

$$\epsilon_q = \frac{S_q}{E} - \mu \frac{S_p}{E}$$

Where μ = Poisson's Ratio

E = Modulus of Elasticity

Solving the strain equations for stresses in the direction of the strains, the following fundamental equations are obtained:



■ Fig. 4 – Delta arrangement of three strain gages on shear panel with stringers

$$S_p = \frac{E}{1-\mu^2}\left(\epsilon_p + \mu\,\epsilon_q\right)$$

$$S_q = \frac{E}{1-\mu^2}\left(\epsilon_q + \mu\,\epsilon_p\right)$$

Therefore, if the strains are known in the direction of the principal stresses, a solution for the principal stresses can be obtained.

In most cases, however, the direction of the principal stresses is not known. The use of three known strains in a direction 60 deg to each other will give all of the principal stresses. The direction in relation to any reference line of these stresses may also be determined.

The stress in any plane is:

$$S_{\theta'} = S_p \cos^2\theta' + S_q \sin^2\theta'$$

$$S'_{\theta'} = S_p \sin^2\theta' + S_q \cos^2\theta'$$

By substituting proper strain values in the above equations, the strain is:

$$\epsilon_{\theta'} = \left(\frac{\epsilon_p+\epsilon_q}{2}\right) + \left(\frac{\epsilon_p-\epsilon_q}{2}\right)\cos 2\theta'$$

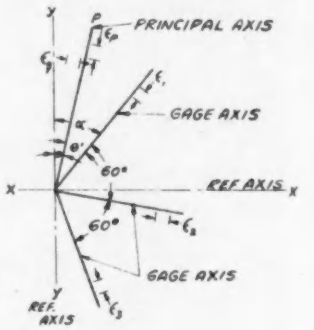
If the terms $\frac{\epsilon_p+\epsilon_q}{2}$ and $\frac{\epsilon_p-\epsilon_q}{2}$ are defined as ϵ and Δ respectively, then the strain in any direction in terms of the principal strain values is $\epsilon_{\theta'} = \epsilon + \Delta\cos 2\theta'$

As shown by the figure, the following equations result:

$$\epsilon_1 = \epsilon + \Delta\cos 2\theta'$$

$$\epsilon_2 = \epsilon + \Delta\cos 2(\theta'+60^\circ)$$

$$\epsilon_3 = \epsilon + \Delta\cos 2(\theta'+120^\circ)$$



The angle α is known from the installation of the gauges. A solution of the above equations gives the following results:

$$\epsilon = \frac{\epsilon_1+\epsilon_2+\epsilon_3}{3}$$

$$\Delta = \frac{\epsilon_1-\epsilon}{\cos 2\theta'}$$

$$\tan 2\theta' = \frac{\sqrt{3}}{1+2\left(\frac{\epsilon_1-\epsilon_2}{\epsilon_2-\epsilon_3}\right)}$$

Since ε and Δ are already defined, a complete solution of the principal strains may be obtained.

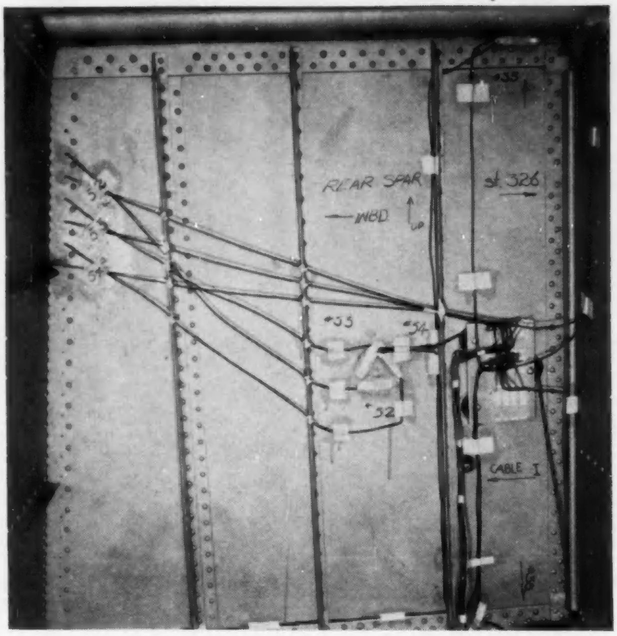
$$\epsilon_P = \epsilon + \Delta,\ \epsilon_q = \epsilon - \Delta,\ \text{And } \theta = \alpha - \theta^1$$

Knowing the principal strains, the principal stresses may be calculated as previously mentioned. The maximum shear stress follows from a known relationship.

$$T_{max.} = \frac{S_P - S_q}{2}$$

In our routine work with this mathematical machinery, we have found it expedient to prepare a standard form on which all the necessary operations are performed, starting with three strain readings and ending with the angle of the principal axis, the maximum and minimum principal stresses, and the resultant shear stresses.

The use of such strain deltas has become standard practice for determining stresses in thin sheet. A typical installation on a spar web is shown by Fig. 5. For the general case in aircraft work, such sheets will be operating in the wave state. The additional bending stresses imposed by the buckling of the sheet are surprisingly large. The effect of such large bending stresses is to render the indications of a delta, attached to one side of the sheet only, very unreliable so far as determining average conditions is concerned. For that reason, unless it is known that buckling will not occur, deltas are invariably installed with gages back to back. Each leg of the delta is then connected with its counterpart on the opposite side of the sheet to read an average stress which automatically cancels out the additional bending stresses due to wave formation. Fig. 6, which is based on tests conducted to investigate these points, illustrates them very well. Note the wide divergence in shear stress indicated by the surface stresses on one side of the sheet only, by comparison either with the delta on the other side or with the known overall shear. The actual shear is precisely checked by the back-to-back arrangement. It is startling, sometimes, to discover how very different local conditions may be from what general theory might indicate. In this case, the reasons for the divergence are, of course, understood. The ability of the gages to indicate local surface conditions has



■ Fig. 5 – Typical installation of strain deltas on spar web

even been used to advantage to check these very stresses in theoretical work being carried out on the development of the tension field theory. Figs. 7 and 8 are photographs of the two sides of a test laboratory set-up of this nature.

There are cases in which such an arrangement is not feasible: for instance, one side of the sheet may form the inner wall of an integral gas tank. It is possible in such a situation to indulge in harmless trickery and install two deltas on the same side of the sheet, oriented so that each leg is parallel to its counterpart, spaced one-half wave length apart, and connected as before, to read an average strain. The wave length can be calculated for a particular stress with reasonable accuracy ahead of time and tests have indicated that the accuracy is not rapidly affected by slight errors in the spacing.

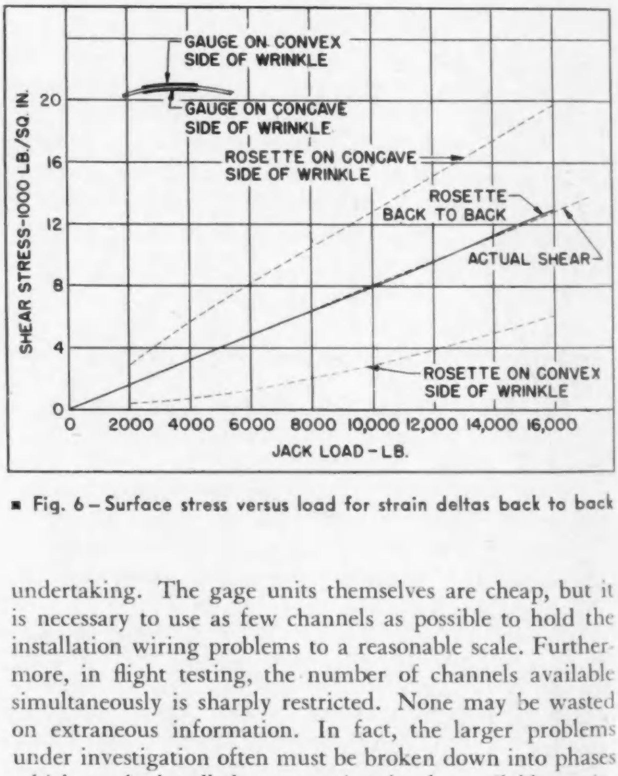
As a further point of interest, while it is, of course, necessary that the orientation of the strain delta be known, tests have substantiated the fact that accuracy is not affected by orientation with respect to the principle stress directions.

There is some gain in accuracy achieved by installing the three gages forming the delta in as small a pattern as possible. A point would be ideal, but unfortunately it is not practicable. The existence of dummy gages for all except total bending differential applications has been mentioned previously. The need for the dummy gage arises from the fact that, in the bridge circuit, it is necessary to balance the resistance of the gage unit itself against an identical resistance. The easiest and most satisfactory way to secure an identical resistance is to use another gage. Furthermore, the most satisfactory temperature compensation is achieved if the extra dummy gage is mounted on metal having the same composition as the metal on which the stresses are being investigated. It is essential, furthermore, that the extra dummy gages be mounted on metal which is not itself stressed. There was, earlier in our work with these gages, a tendency to survey the surrounding structure and innocently mount the dummy gages on any likely looking bit of metal that we hoped might be unstressed. That practice invariably led to difficulty and had to be utterly abandoned. You will notice, in the photograph of Fig. 5 in which the dummy gages for the strain delta are shown, that these gages are mounted on a separate piece of metal so supported that stresses cannot be transmitted to it from the web on which it is supported. Such is the typical and essential practice.

■ Instrumentation Summary

In summary, we have at our command distinct methods for coping with each type of stress with which we may be concerned. Strain, deflection, and vibration are seen to be brought closer together than ever by virtue of the fact that they can be investigated (simultaneously, if need be) by the same instrumentation.

In what follows, reference is made particularly to the use of the methods under discussion, to static tests of complete airplanes, often very large ones, or to tests made during flight. In such cases, the wiring between the gage units and the associated apparatus becomes a considerable undertaking. The gage units themselves are cheap, but it is necessary to use as few channels as possible to hold the installation wiring problems to a reasonable scale. Furthermore, in flight testing, the number of channels available simultaneously is sharply restricted. None may be wasted on extraneous information. In fact, the larger problems under investigation often must be broken down into phases which can be handled one at a time by the available equipment. Means must then be provided to switch from one phase to another when, as is usually the case, more than one phase must be covered during a single flight.



■ Fig. 6 – Surface stress versus load for strain deltas back to back

Fig. 9 shows photographs of the installation of the amplifying and recording apparatus set up during a static test of a complete airplane. In this case, cables and all the wiring from all the gages installed all over the airplane have been brought to a stand conveniently set up outside of the airplane. The unit to the left in the upper left-hand photograph is a six-channel amplifier with its associated

meters. The center instrument is a Brown Electronic Potentiometer being used to record indications from some 300 strain-gage installations. This instrument incorporates an electronic amplifier and a high-speed self-balancing bridge system. A switch panel is shown to the right which permits selection of any one of the 300 channels. For flight-test work, an installation similar to that shown by Figs. 10 and 11 is typical. The rapidly varying nature of the stresses, particularly the short duration of the peak stresses, in which we are most interested, makes it necessary to record the results with some form of recording oscillograph. In our work, a twelve-channel Miller oscillograph is used. In this case, a twelve-channel amplifier and a twelve-channel Miller oscillograph are shown installed in the XB-19, this equipment being thoroughly suitable for dynamic loads in contradistinction to the two preceding illustrations which are of an installation suitable only for static test work.

It is necessary to diverge from discussion of strain gages *per se* for a moment to point out that various other uses may be made of this equipment simultaneously; for instance, very reliable accelerometer pickups, which function as an inductance bridge just as the strain gages function as resistance bridges, have been developed and may be used simultaneously wth the strain gages. Also, both the accelerometers and the strain-gage units are ideal vibration pickups, and they interchangeably and simultaneously record vibration frequencies and their relative amplitudes. Moreover, a little ingenuity will make the same instrumentation supply much additional essential information. For instance, such things may be done as recording pressure variations in a hydraulic line thanks to its expansion and contraction as picked up by a strain gage cemented around the line, or indicating a landing wheel rotational speed by suitable contact points on the wheel, superimposing a record on a channel already being used for some other problem as in Fig. 3. Spatial deflections may be measured by suitably forcing the deflection of a simple cantilever beam which is equipped with bending strain gages and is calibrated in terms of deflection of the beam.

In short, the most effective use of the available channels involves the most suitable combination of strain-gage data with such auxiliary information as may be necessary.

The record from the oscillograph may, therefore, represent an incredible amount of information, often not obtainable in any other way. Utilizing the foregoing techniques, some typical problems to which these methods may be applied will be considered.

■ Applications

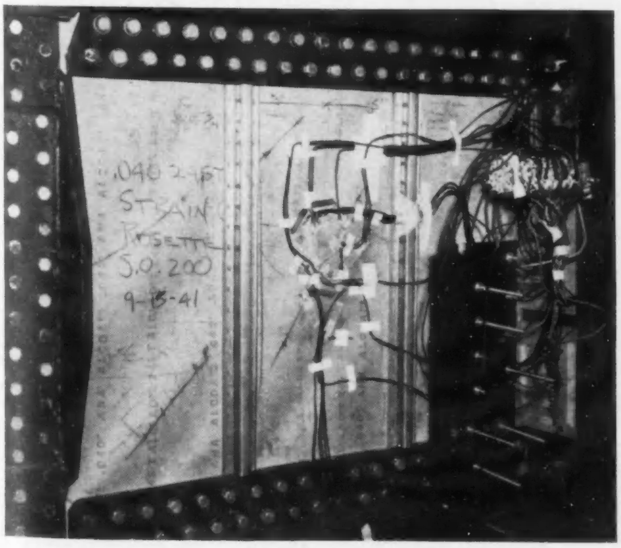
As an example, consider the familiar and still difficult problem of the distribution of direct and torsional shear stresses in a multispar stressed skin wing. The distribution of shear stress intensity is represented diagrammatically by Fig. 12a.

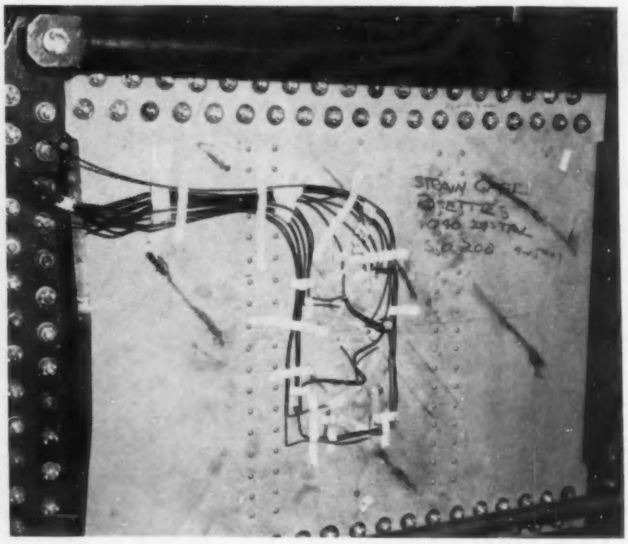
The stresses shown will be the combination of direct and torsional shears plotted vertically from the contour of the wing section as a base. Their value largely determines the skin thicknesses, the spar web thicknesses, something of the stiffener arrangements involved, and the rivet patterns for seams in the wing covering. Therefore, the accurate evaluation of these intensities is important. The methods of calculating such shear intensities are somewhat involved. A substantiation of their accuracy is, of course, desirable.

From what already has been said it is obvious that, at a particular cross-section of the wing, strain deltas could be installed at each point of maximum shear stress intensity, both on the wing covering and the spar webs. Their arrangement is indicated by Fig. 12b. For a wing such as the one sketched, this installation would involve 12 deltas, or 36 channels. So great a number of channels limits such an investigation to static test work rather than flight test.

In calculating such a stress distribution, the usual procedure is to distribute the direct shear among the spars in proportion to the spar web area. Such a procedure is based on the fact that the twisting deflections of a monocoque wing are small.

The air load applied to the wing and acting at its center of pressure causes a torsional moment about a spanwise



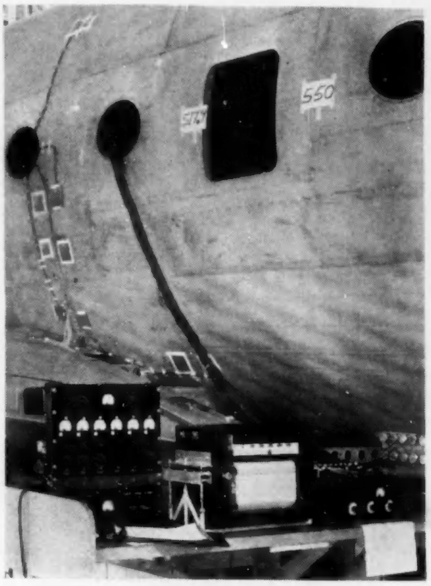


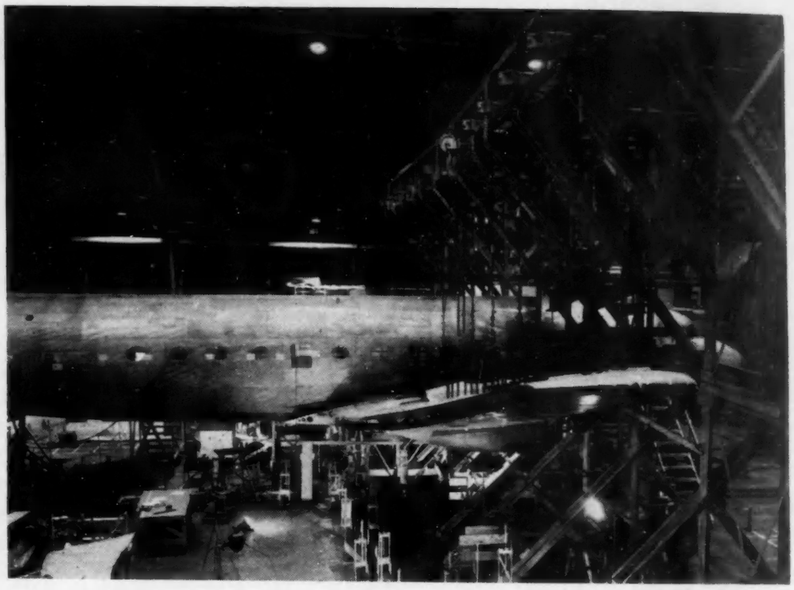
■ Figs. 7 and 8 – Two sides of test laboratory set-up showing back-to-back arrangement of strain-gage "rosettes"

axis of the wing. The magnitude of this moment depends on the distance of the selected axis from the air load center of pressure. For convenience, the elastic axis, defined as the locus of points about which no angular deflection would occur were load applied at that point, is the usual reference axis. The corresponding torsional moment is then distributed among the various torque boxes (three in Fig. 12) in proportion to the torsional rigidity of each. Finally, the shear stresses in the wing covering involved in carrying the bending loads to any bending material located between the shear webs, are calculated. All three shear stresses are combined to give the typical pattern of shear-stress intensity of Fig. 12a. This procedure, therefore, involves the determination of the elastic axis position in order to establish the value of the torsional moment.

So far, agreement between mathematical methods of calculating the elastic axis position with test determination has not been satisfactory.

Since the axis cannot be located accurately during the design stage, common practice has been to define conservatively a range of chord positions within which the axis should lie and to design the wing structure for the loads implied by the extremities of the assumed range. This method involves a doubling of much analysis work





■ Fig. 9 –Installation of the amplifying and recording strain-gage apparatus set up during a static test of a complete airplane

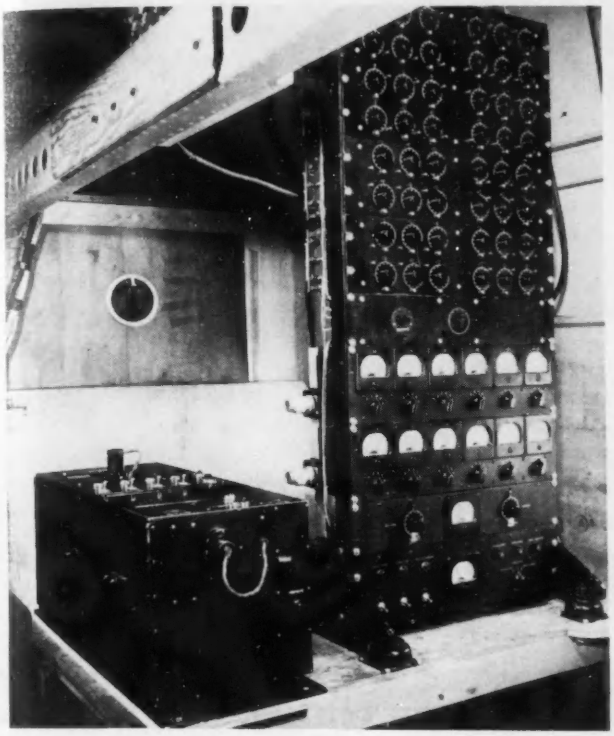


Fig. 10 – Front view

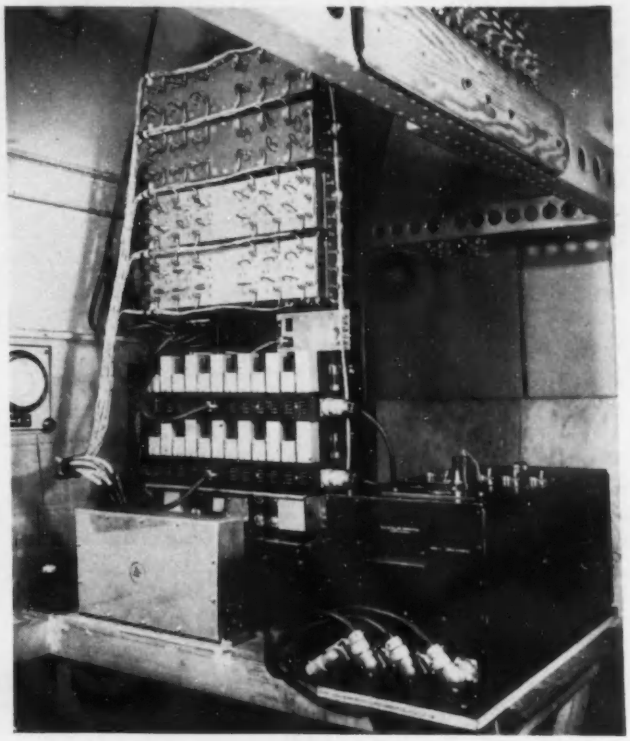


Fig. 11 – Rear view

■ Figs. 10 and 11 – Typical installation used for flight-test work

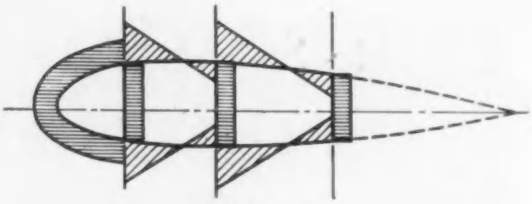
and the inclusion of more structural weight than is actually necessary were the position of the elastic axis accurately known.

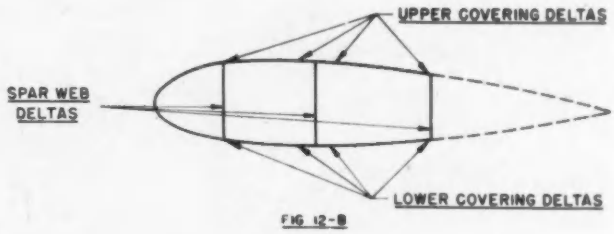
Starting from the strain-gage data derived from an installation as indicated by Fig. 12b, it is possible to work in the opposite direction, separate out the various contributory shear stresses, and arrive at the true shear center or elastic axis. That is being done, and a wider experience with it will lead ultimately to reduced analysis time and structural weight.

It may be noted in addition that a similar procedure would be very suitable for an investigation of the shear lag problem arising from the shear deflections which correspond to the shear stresses in the wing covering. The ability to determine experimentally the actual shear distribution by use of a shear delta on the wing covering between each item of bending material should accelerate greatly the development of a mathematical treatment suitable for several engineering applications.

As another sidelight on the use of a similar installation in flight-test work, there recently has been a growing concern over the effects of compressibility upon such items as the aerodynamic twisting moment imposed on the wing at velocities approaching the speed of sound. The increases predicted by pure theory have not been evident in flight-test work so far conducted. Actual measurements of the magnitude and, more particularly, of the variation in magnitude with dive speed from a simplified version of such an installation are planned to throw light on this subject.

Another basic problem is the substantiation of the broad assumption that the simple bending theory applies to a





■ Fig. 12A (above) – Distribution of shear-stress intensity in a multi-spar stressed-skin wing

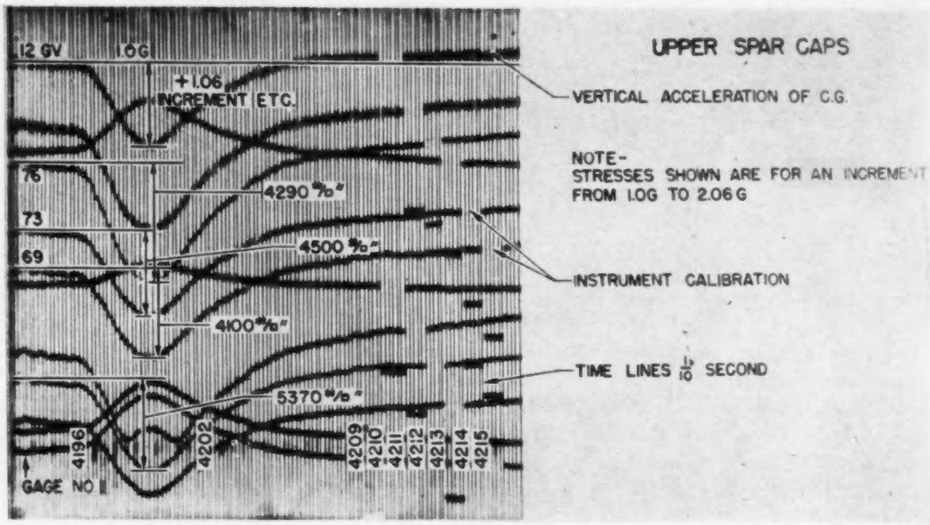
■ Fig. 12B (below) – Arrangement of strain deltas at points of maximum shear stress intensity at a particular cross-section of wing

beam of the complicated cross-section of a typical multi-spar cantilever wing. I have mentioned previously the usual assumption that twisting deflections are small, which is to say that each spar assumes the same deflection curve. Presumably, spars are forced to the identical wing deflection curve (as distinct from their individual curves) by the torsional rigidity of the outer covering and the stiffness of the ribs. Actually, neither of these considerations is infinitely effective, and some independent motion of spars does occur. Such independent spar action is only likely to become important where special design problems occur such as discontinuous spar arrangements to provide room for retractable landing gears or opening for bomb bays, and so on. In such cases, the design problem may become very difficult indeed.

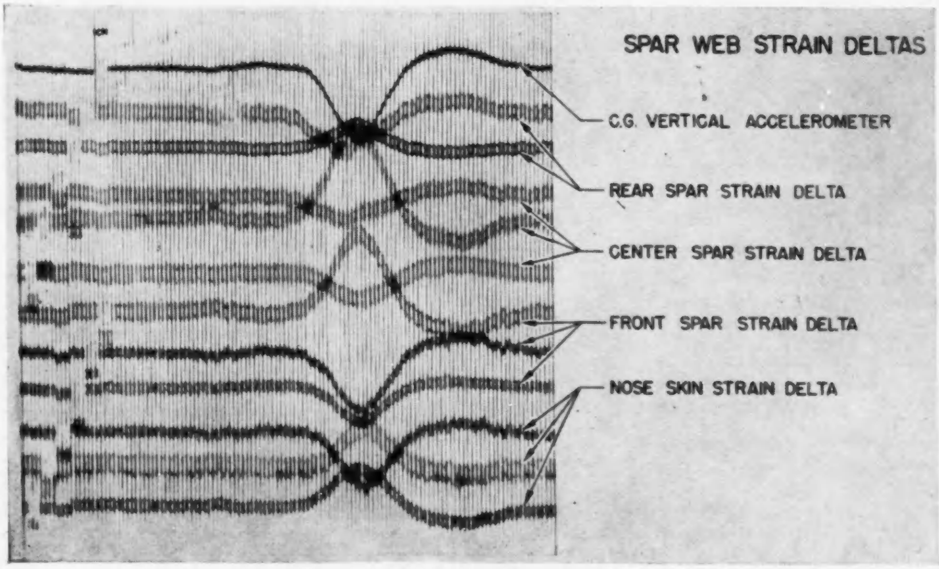
Assumptions as to the behavior of the spars under load must be made, and the resulting structure itself becomes a function of those assumptions. Substantiation of the assumptions that have been made in a particular design are therefore important, both for that design and for the development of good judgment in assumptions that must be made for new designs.

At least one very complicated problem of this nature has been investigated in flight. For that purpose, the state of stress of several critical cross-sections of the wing in the affected area was determined by data from average reading strain-gage installations on each spar cap and strain deltas on each spar web. By comparison with calculated stresses and readjustment of the original design assumptions, a better understanding of the behavior of the entire assembly is achieved. This work on an experimental airplane means that a production redesign for the usual conditions of greater power and greater weight would be accomplished with more dispatch and precision than would otherwise be possible. This might effect a long-range saving many times greater than the cost of securing the information.

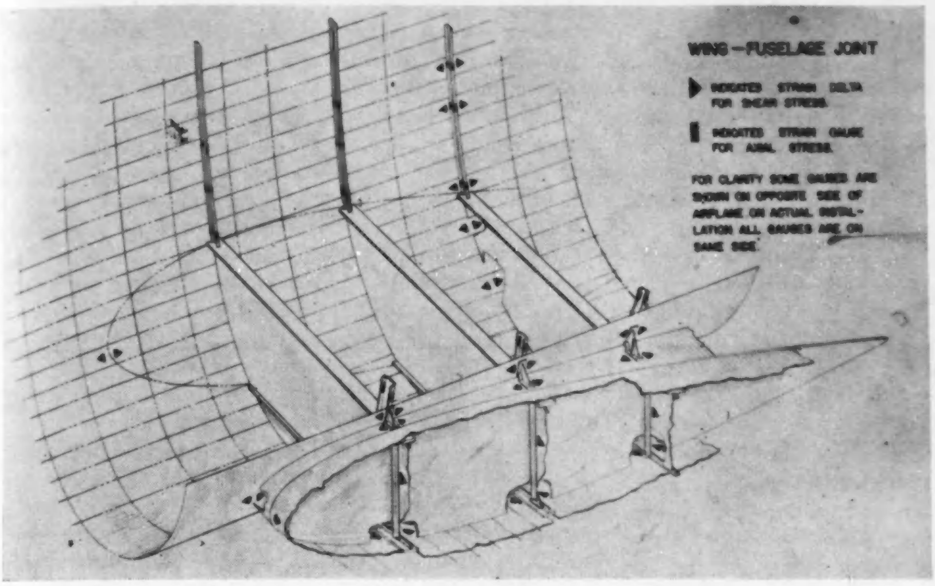
Fig. 13 is a reproduction of a portion of the records obtained during this particular investigation. The traces showing the variation of stress in the upper spar caps at one section of the wing have been marked, together with some auxiliary data. The reproduced portion of the record is typical. From it, the entire history of events throughout a sharp pull-up, one of many made for these test purposes, is determined. The magnitude of the stress at any point



■ Fig. 13 – Record obtained in flight investigation of stresses in critical cross-sections of wing



■ Fig. 14 – Another typical record of strain deltas on spar webs in flight

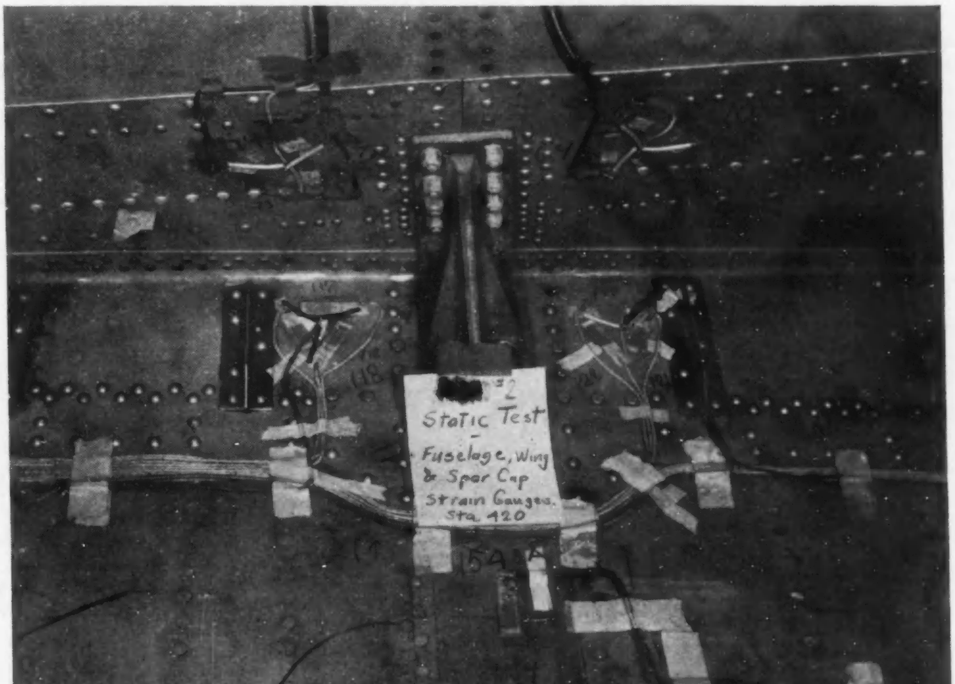


■ Fig. 15—Arrangement of strain gages and strain deltas for stress investigation of wing-fuselage joint

and at any instant is indicated by the departure of the trace from its steady flight condition base. Fig. 14 is another part of the data typical of records of strain deltas on the spar webs under the same conditions.

One of the basic problems with which the structural designer is always faced in proportioning the structure of a new design is the nature of the transfer of load between the wing and fuselage at their intersection. In large all-metal designs, for reasons of structural continuity, there is usually, in addition to fittings between spars and fuselage frames, a continuous shear connection between wing covering and fuselage skin. If it were not for this shear connection, the magnitude of the reactions at the fittings would be a simple problem in statics. With it, however, the state of affairs is indeterminate.

The usual procedure in design is to make overlapping assumptions as to the proportion of the torque load that is carried directly by the wing to fuselage fittings and as to the proportion that is carried by direct shear transfer from the wing covering to the fuselage skin. The problem is of particular importance whenever, for functional reasons, it is desirable to keep the interior fuselage space free of obstructing bulkheads. This arrangement means that loads transferred to the fuselage by the fittings are actually carried by ring frames which are usually shallower in depth than the structural designer would prefer.



■ Fig. 16—Part of actual installation for stress investigation of wing-fuselage joint

Precise knowledge of the actual nature of the load transfer, therefore, determines the general proportions of the wing-to-fuselage fittings, the spar frames, the fuselage skin in the neighborhood of the wing intersection, and the rivet patterns in that area. The problem is, furthermore, usually complicated by the existence of windows, doors, and emergency hatches in the affected part of the fuselage.

While we have come laboriously over a period of years to have a fair understanding of the state of affairs, we still must resort to overlapping assumptions, which phrase is a reassuring way of saying that the design is overly conservative.

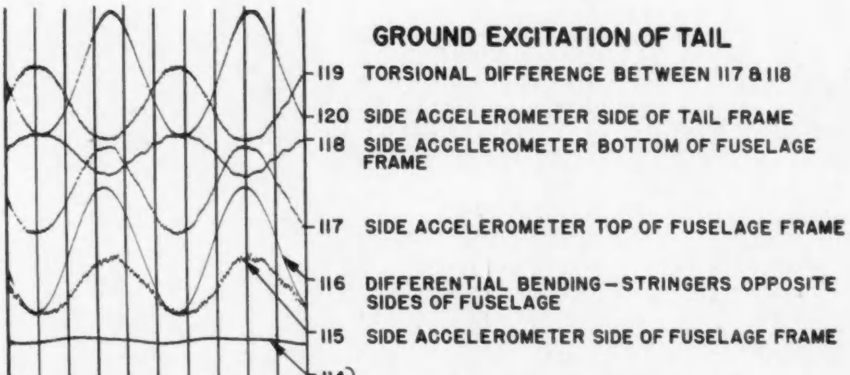
Part of the investigation being covered by the set-up shown in Fig. 9 was concerned with this problem, shown diagrammatically by Fig. 15. Strain deltas at critical points in the wing shear system just outboard of the fuselage, on the fuselage skin adjacent to the wing intersection, and close to the spar frames, together with a number of gages sufficient to determine loads being imposed on the spar frames immediately above the wing fittings, permit a precise determination of the nature of the load transfer. Fig. 16 is a photograph of a part of the actual installation. It is recognizable by comparison with Fig. 15 as that region on the upper wing surface at the root of the center spar.

The resulting information, like all similar information, serves to substantiate the design assumptions that have been made; permits more intelligent revision of design as inevitable increases in load-carrying capacity during the future of the particular design is concerned; permits more accurate disposition of shop salvage problems that may arise in the affected area; and, last but not least, permits the reduction, or perhaps the elimination, of the need for overlapping assumptions in the design of the next similar airplane.

It has been previously mentioned that both the strain-gage and accelerometer pickups were suitable for vibration recording work. In fact, they become inseparable in flight-test work for every strain recording channel records any variation in stress that may result from oscillations in the aircraft structure. That feature is sometimes taken advantage of in order to determine the nature and the seriousness of such oscillations as may occur. For instance, Fig. 17 is a section of the oscillograph record from an installation that was set up for the express purpose of determining the severity and structural importance of the mild oscillation of the tail that was known to occur in an airplane under investigation. The tail was excited at the frequency with which we were concerned on the ground, and the record of Fig. 17 was made. It was obvious from observation that the amplitude of the oscillations was much greater on the ground than in flight; nevertheless, the stresses so measured were determined and were found to be negligible. An identical record was then made with the airplane in flight while similar oscillations were occurring, and it was possible to demonstrate that the oscillations occurring in flight were much less serious. It was shown also that the stresses resulting from flight oscillations were correspondingly less serious than those which had occurred on the ground, which were, as we have seen, negligible. This was an important point, and one that it might not have been possible to establish so completely by any other means.

■ Limitations

So far, I have dealt with the favorable side of the picture. I have discussed what we can and have done and, in addition, I have tried to give some idea of the nature of the problems to which the application of equipment of this type is suitable. In order to restore a proper sense of proportion, however, it will be necessary to discuss some of the difficulties that are involved. There are, of course, technical difficulties. Low stresses may be indicated if there exists a lag due to imperfect cementing of the gage to the surface in question. Yield of the wire of the gage itself at higher stresses may cause difficulties; moreover, troubles common to any involved electrical circuit may be encountered, and so on, in an ever-increasing list. Such problems, however, properly belong to a discussion of the development of the equipment itself rather than to its



■ Fig. 17 – Section of oscillograph record from investigation of oscillation of tail

application, and reference is made again to the same two papers mentioned earlier. It is not the intention to imply that the development of equipment is completed. There are many problems yet to be solved.

Aside from technical difficulties with the equipment itself, there are endless human problems involved in any large-scale application of this technique. One definitely cannot expect the mere thoughtless application of huge numbers of gages to lead automatically to wonderful results. Careful thought and planning ahead of any investigation result in reducing the complication of the installation to a minimum and are utterly essential. Preparation for the rapid, accurate, and permanent recording of pertinent data is necessary. Organization must be provided to assure rapid reduction of data once recorded and, last but not least, an almost unlimited degree of patience is required.

Such difficulties as do occur may not be so serious in laboratory work for tests can be repeated there with reasonable ease but, in large-scale static test work and in flight-test work, an error that invalidates a recording of events may spoil an opportunity to secure data that will never occur again. In flight-test work, in particular, in allotting flight time to this phase of the work, it is very wise to anticipate repetition of each run several times until the entire installation is known to be in perfect order. Such preliminary trials usually will be necessary before a satisfactory record is obtained.

Even apparently satisfactory results, so far as the equipment is concerned, have to be used with a great deal of understanding. We know from sad experience that we have to be smarter than the gages in every application of them. They will give false answers and misleading information to anyone who will believe too easily. It is necessary to have sufficient understanding to recognize misinformation, to analyze the nature of it, to trace the trouble to its source, and to repeat that process until satisfactory results are being achieved. It is heartbreaking, after days and weeks of work and planning, to reduce the results contained in a record that appears satisfactory only to discover that some vital part of the installation is not working as intended. Such occurrences always seem to strike at the one most vital point. I do not know whom I quote, but whoever it was that first referred to the "innate perversity of inanimate objects" could well have been discussing his experience with such equipment.

No one should feel that the application of this technique is cheap, or that it is easy – or even that one will always like the significance of what may be discovered. There is the further point that, while we are rapidly solving problems that have puzzled us for years, we are with equal rapidity opening new problems. That is to be expected. The net result, however, is that, while our product is being improved at an accelerated rate, our problems seem more numerous than ever.

The most serious problem of all, under the present circumstances, may well be that of limiting these methods to applications reasonably sure to produce directly useful information and not a confusion of unassimilated data.

I have purposely refrained throughout this paper from too great detail or actual figures which would apply only to specific cases. I have been primarily concerned with conveying the idea that there are techniques and equipment available, the application of which, though difficult, gives an insight into the problems of structural behavior, which has never been possible before. Such an insight is particularly important in aircraft work with thin metal structures, the whole theory of which is still in the development stage. It is hoped that the end result is a better and a simpler structure, hence, an airplane that is easier to build and more efficient when, at last, it is put to use.

Controllable Pitch Propellers for Light Airplanes

THE subject of controllable pitch propellers for the light commercial airplane has been the serious concern of many designers for several years. The need of such a propeller was probably first seriously felt by designers of low horsepower multi-engine airplanes. The comparatively recent development of high performance light planes has further extended this need until at present it approaches a necessity or a bottleneck restricting further progress in airplane design of low horsepower.

The first step toward the realization of a propeller of this type was the development of a satisfactory adjustable pitch propeller of moderately low price. Since our company introduced the Freedman-Burnham adjustable-pitch propeller at the Chicago Air Show in 1937, a study has been going forward in an effort to produce a suitable controllable pitch hub to be interchangeable with the standard adjustable pitch hub, using the Freedman-Burnham wood and plastic propeller blades. In connection with this study we have furnished technical assistance and otherwise encouraged several inventors and designers who presented controllable propeller hubs designs for consideration.

■ Multi-Engined Aircraft

The development of low-powered multi-engined aircraft, especially the two-engine type, has been definitely retarded because of the lack of a controllable propeller.

With this type airplane all of the performance requirements of the single-engine airplane must be executed with one engine inoperative. It is a difficult problem to design a twin-engine airplane with a satisfactory single-engine performance of any horsepower. The present high performance transport and service airplanes of this type would be impossible without the benefits of the modern controllable pitch propeller.

The problem of single-engine performance of the light twin-engine airplane is more difficult than that of the higher powered craft because of the adverse effect on the ratio of power available to power required in the case of the low powered craft.

For installation on a twin-engine aircraft, the maximum benefit is realized if the controllable propeller is of the full feathering type.

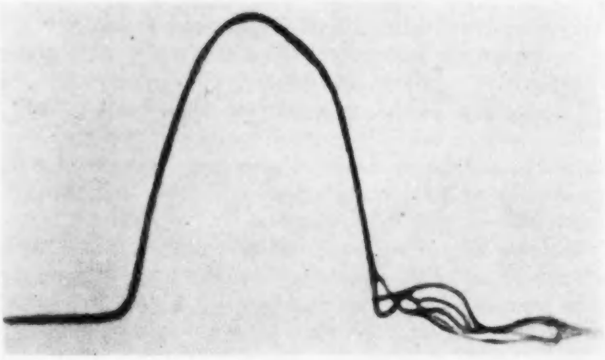
In conclusion, the controllable pitch or constant speed propeller opens the door to advances in airplane and engine design which are definitely closed if only a fixed pitch propeller is available. Furthermore, the ultimate gain in performance from such an arrangement will come only when the airplane and engine are designed to take advantage of its possibilities.

Excerpts from the paper of the same title by Walter E. Burnham, chief engineer, Freedman-Burnham Engineering Corp., presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 13, 1942.

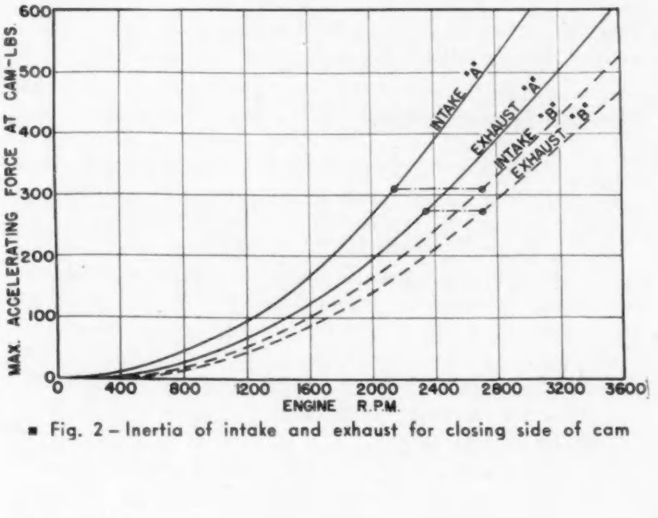
Some Results of
As Applied

by CARL VOORHIES

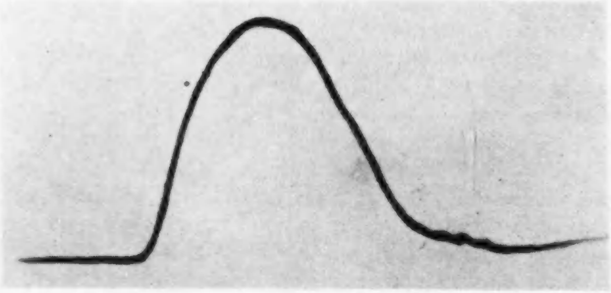
Wilcox-Rich Division, Eaton Mfg. Co.



■ Fig. 1 – Typical condition of false motion of the valve gear



■ Fig. 2 – Inertia of intake and exhaust for closing side of cam



■ Fig. 3 – Performance of valve gear shown in Fig. 1 at same speed but with cam redesigned to give higher opening and lower closing accelerations

IN designing a poppet-valve engine, the entire valve gear, from the camshaft to the valve seat, must be considered as a whole, and it is impossible to state what the design of one part should be without relating it to all the other parts. For instance, the rocker ratio would greatly affect the cam section and pushrod; the valve guide would have a bearing on the cam contour design.

It is therefore the purpose of this paper to touch upon the mechanical design of each component of the valve gear, starting with the camshaft and continuing up through the tappet, pushrod, and so on, to the valve-seat insert. We shall conclude by devoting some time to a consideration of hydraulic valve lifters.

There is in progress a continually increasing demand for speed in diesel engines, in order to develop more and more power from a given size engine. This trend necessitates operating valve gears at higher speeds, and also, increased rates of valve opening and closing. This condition is further complicated by the necessarily heavy reciprocating parts.

■ Cam Design

As we increase engine speeds, the valve gear becomes more in evidence by the trouble it can cause, especially in cases where provisions are not made in the design to turn to higher speeds.

False motion may become excessive, causing breakage of parts, pounding-in of valve seats, stretched valve stems and dished heads, and tappet face or cam failure – all to such an extent that it becomes necessary to redesign the valve gear for operation at higher speed.

It has been found that false motion of the type that causes this trouble is due to the flexibility of the valve gear itself, storing energy during acceleration and releasing it to cause a bouncing motion of the valve gear, which may be increased in amplitude by the vibrations of the valve spring.

A typical condition of false motion is shown in Fig. 1. At first the valve follows a path corresponding to the lowest curve; the amplitude of the bounce gradually builds up until the valve strikes through the ramp of the cam, and contacts the seat at high velocity; it then bounces on a path as shown by the uppermost curve, at which point the phase is broken up and repetition of the cycle commences.

This type of false motion is a function of acceleration loads, and the maximum permissible acceleration is de-

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VALVE-GEAR RESEARCH to Diesel Engines

termined by the rigidity of the valve gear. Where reciprocating parts are rigid, and where cam bearings are close together, and the body of the shaft is of sufficient diameter to minimize deflections, a relatively high acceleration can be used.

Since acceleration loads are the limiting factor in the speed of a valve gear, the maximum speed of any given harmonics of the valve spring which obviously become worse as speeds increase. In a great many cases it has been possible to correct trouble in valve gear by determining the maximum cam accelerations at which the valve gear will operate satisfactorily, then designing a cam which does not exceed this acceleration at the desired engine speed. It also is sometimes possible to increase perform-

EMPHASIZING first that the entire valve gear from the camshaft to the valve seat must be considered as a whole in designing a poppet-valve engine, Mr. Voorhies proceeds in this paper to discuss the mechanical design of each component of the valve gear. Considerable space in this discussion is devoted to the design, operation, and advantages of hydraulic valve lifters.

A summary of the various points set forth in this paper includes the following recommendations:

Camshaft deflection should be held to not more than 0.002 to 0.003 in. Accelerations should be as high as possible on both the opening and closing sides; the opening side acceleration may range up to three times that of the closing side. Cam followers should not be overstressed by the use of too small a nose radius. Mushroom-type followers may have a cast-iron face when operating on cams of any material. Steel followers can be used successfully only on cast-iron cams provided that the cam is properly designed; the section of the cast-iron cam should be about 30% greater than that of a steel cam for equal rigidity.

The pushrod should be of tubular welded-end construction. The rocker arm should be of the lightest construction that will not deflect excessively under maximum load; a rocker ratio of 1.3 to 1.6 is best, considering all factors. The use of a roller is very good practice in large engines where loads are high. Rocker-arm geometry should be such that the movement of the valve is about one-third above and two-thirds below the centerline.

The valve should have a hard tip, high hot strength, and stainless properties. For relatively high speeds a hollow-drilled valve is desirable for lightness. If this design causes excessive heating, sodium cooling should be used.

Valve stem guides should be extended as high as possible on the valve to permit ample clearance without excessive valve cocking.

The spring should have a safety margin of at least 30% at the point of reversal, and should have the highest possible frequency without overstressing the material.

The valve seat insert should be of ample section, maximum rigidity, and of a material which provides maximum resistance to distortion.

■ ■ ■

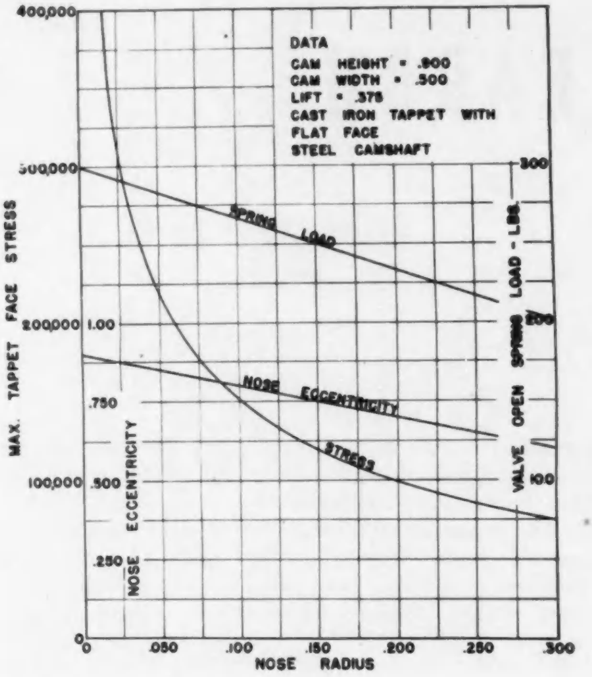
THE AUTHOR: CARL VOORHIES (M '27), research engineer, Wilcox-Rich Division, Eaton Mfg. Co., has been with that company for more than 10 years, specializing in valve gear development. Beginning with Duesenberg in 1922, Mr. Voorhies' wide experience in automotive engineering includes work on passenger cars, racing cars, and trucks.

valve gear can be controlled by designing a new cam contour, once the maximum permissible acceleration is found.

Referring to Fig. 2, an engine having a camshaft with an acceleration load as shown on the upper curves, can be turned to higher speeds by providing a camshaft having an acceleration represented by the lower curves. The tolerable accelerating load seems to be a straight horizontal line with slight variations, which can be attributed to the

ance of engines by raising the acceleration to the maximum that can be used.

In designing a cam to reduce false motion, one must first consider the fact that, when a valve bounces on the opening side of the lift, it is bouncing in the general direction in which motion is desired, whereas on the closing side this is not the case. This condition means that any valve gear can stand considerably higher accelerations on



■ Fig. 4 – Cam nose radius versus maximum tappet face stress, nose eccentricity, and spring load

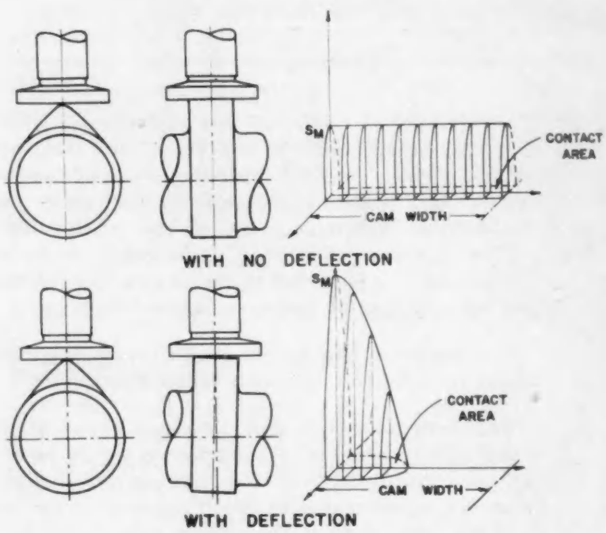
the opening side than on the closing side for a given amount of false motion. We have been using this information in cam design for some time, but had not realized the extent to which this could be done to advantage until we recently made an investigation using a cathode-ray oscillograph on an overhead-valve engine. The first cam design for this engine had accelerations at 4000 rpm of 7220 ft per sec^2 on the opening side, and 4405 ft per sec^2 on the closing side. The lift curve of this cam at 4750 rpm is shown in Fig. 1. It will be noted that, in spite of the high opening acceleration, there is no bouncing. Yet, with the lower acceleration on the closing side, considerable bouncing is indicated. From these results, it was decided further to reduce closing acceleration and, in order to accomplish this purpose, it was desirable to increase opening acceleration. A cam was therefore designed having accelerations at 4000 rpm of 9845 ft per sec^2 opening, and 3080 ft per sec^2 closing. Valve-gear performance with this cam at the same speed is shown in Fig. 3, which indicates that there is still very little false motion on the opening side, while at the same time, valve bounce on the closing side is reduced considerably.

One of the advantages of this type of cam is that the lift at the point of maximum piston velocity is increased over the conventional symmetrical cam with the same lift and lift area. In the case of these particular cams, the later design showed a maximum power gain of 4%. This increase can be attributed partly to lift-curve characteristics and partly to valve-gear behavior.

■ Tappet Face Stress

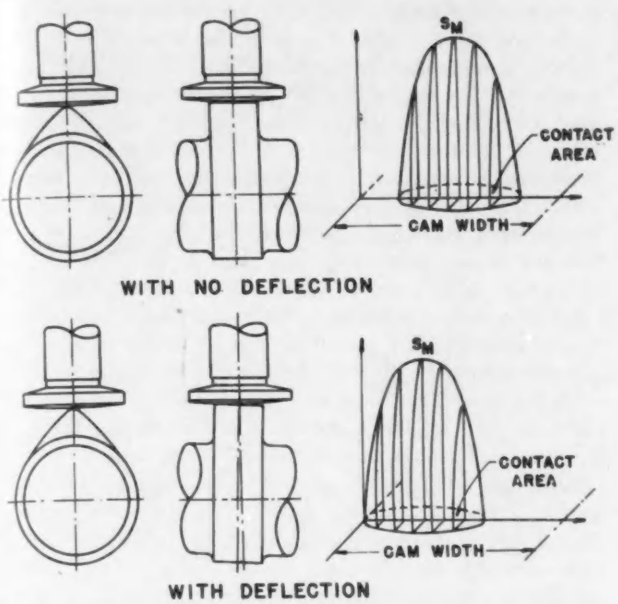
Another item which must be considered in camshaft design is the stress of the follower and cam at their point of contact. The deceleration rate is increased as the nose radius is decreased. Therefore, referring to the chart in Fig. 4, it will be seen that the stresses at the nose are increased due not only to the smaller contact area but also to the higher spring loading required by the higher decelerating load. From records kept for some time of the stresses between the cam and the follower, at both the point of maximum acceleration and on the nose of the cam, it has been found that failure accompanies high stresses on the nose and apparently has no connection with the stress at maximum acceleration. In fact, the stress at maximum acceleration is usually low where failure occurs, because the high deceleration and the high stress on the nose are necessarily a result of a relatively low acceleration for a given valve lift.

A method has been developed of determining the stresses between the cam and follower by adapting the Herz equation. This consists of calculating the highest stress over an area of contact – this area being determined from the modulus of elasticity and the contour of the materials. In the case of two cylindrical radii in contact, the area is a rectangle in which the stress is at a maximum on a line through the middle and tapers off to zero at the edges. Where a spherical radius contacts a cylindrical radius, the area is an ellipse with the highest stress at the center point.



■ Fig. 5 – Contact area and stress between tappet and cam represented by three-dimensional diagrams – flat-face tappet with no deflection and with deflection

Fig. 5 represents the contact area and the stress between tappet and cam by means of three-dimensional diagrams. These views show areas of contact and stress between cam and a flat follower. At the top is represented the theoretical condition, while the diagram at the bottom illustrates what may occur when any misalignment exists between cam and follower. A very little misalignment may cause stresses two or three times the theoretical. Field results indicate that, where flat followers are used, 121,000 psi is the lowest stress where failure occurs, and 125,000 psi is the highest stress operated successfully.



■ Fig. 6 – Contact area and stress between tappet and cam represented by three-dimensional diagram – spherical-face tappet with no deflection and with deflection

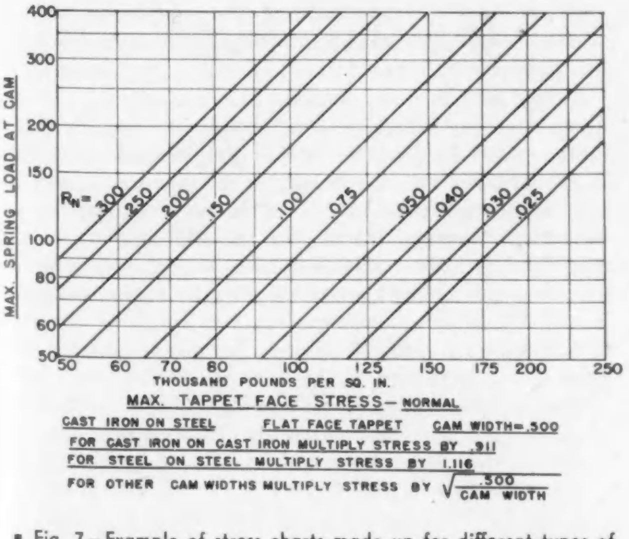
Fig. 6 illustrates, in the same manner, the contact area and the normal stress where a spherical radius contacts the cam. It is obvious that, in the event of misalignment with the face of the cam, there is no change in either the contact area or the stress as long as the cam has sufficient width to keep the full pattern of the bearing on the face of the cam. Field results indicate that cast iron with spherical face on steel has a limit of 189,000 psi maximum stress and that there is very little variation in the point of failure. Fig. 7 is an example of stress charts made up for the different types of followers. Due to the difference in the modulus of elasticity, it is necessary to consider this relation separately for different combinations of tappet and cam materials. It is also necessary to consider different cam widths for flat-faced tappets. The chart is drawn on the basis of a cast-iron, flat-faced tappet on a steel cam, ½ in. wide. A conversion factor of 0.911 is used for obtaining the stress of cast iron on cast iron, and a conversion factor of 1.116 for steel on steel. For cam widths other than ½ in., the factor would be the square root of 0.500 in. over the cam width. Reading the chart, the spring load is given as the ordinate, and the stress as the abscissa. The almost diagonal lines are the lines of constant nose radius, and the stress is determined on the abscissa from the point of intersection between the nose radius and the spring load.

The stress chart for a 30-in. spherical radius tappet is shown in Fig. 8. Here we use conversion factors for different material combinations, but none is required for cam width as it is assumed that the cam is wide enough to prevent the bearing pattern from extending over the edge.

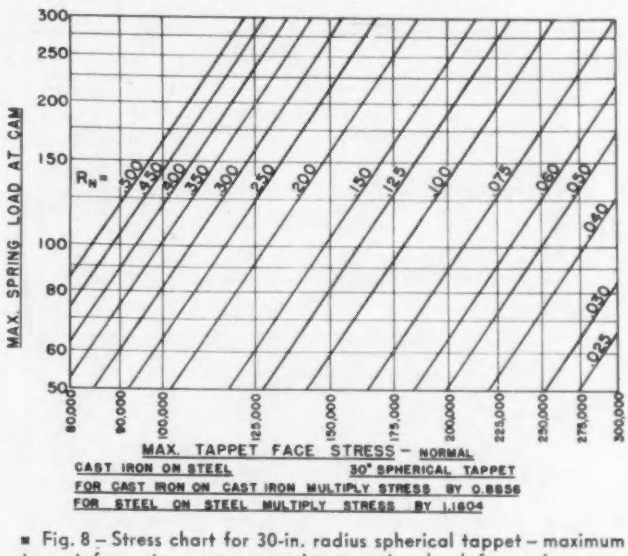
Fig. 9 is a similar chart for a 100-in. radius tappet.

Consideration of the conversion factors used with the charts indicates that the best material combination is a cast-iron tappet operating on a cast-iron cam. Actually, however, steel on cast iron has run with a higher stress than any other combination, the limit being approximately 214,000 psi. The limit for cast iron on cast iron is approximately 175,000 psi. Steel followers can be used successfully only on a cast-iron cam. Generally speaking, whenever a steel mushroom follower operates without failure on a steel cam, it is an indication that the cam could be improved with regard to acceleration, deceleration, and timing, to give increased engine performance. The steel-on-cast-iron combination is sometimes desirable from the standpoint of reciprocating weight but, for equal rigidity, a cast iron-shaft should be approximately 30% greater in section than the steel shaft.

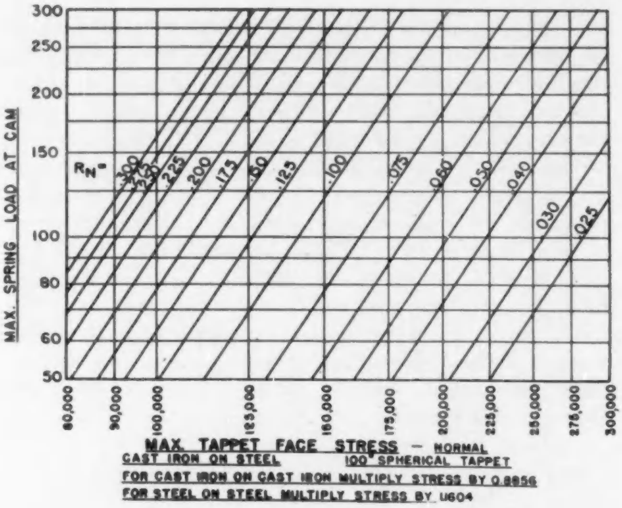
Fig. 10 shows a comparison of stress, based on a



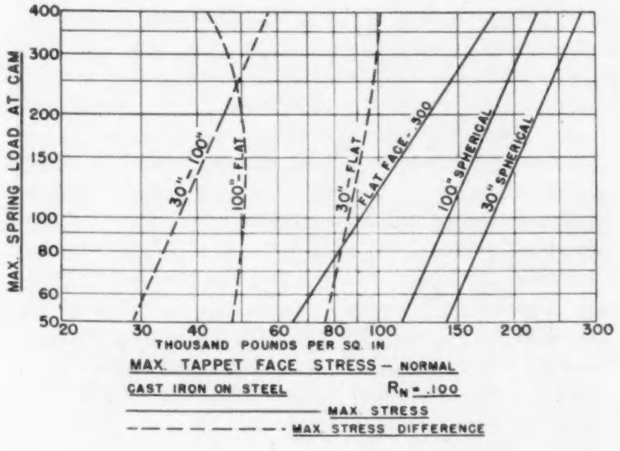
■ Fig. 7 – Example of stress charts made up for different types of followers – maximum tappet face stress versus maximum spring load for various cam nose radii



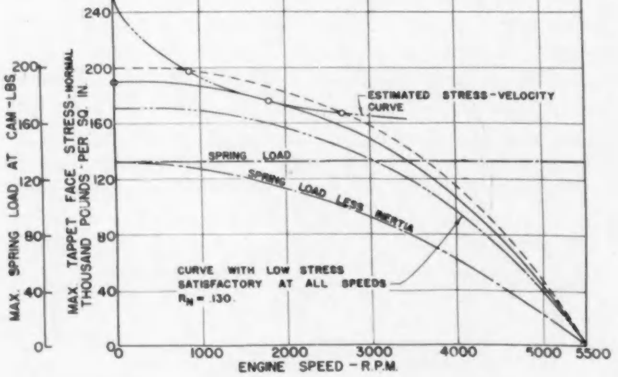
■ Fig. 8 – Stress chart for 30-in. radius spherical tappet – maximum tappet face stress versus maximum spring load for various cam nose radii



■ Fig. 9 – Stress chart for 100-in. spherical tappet – maximum tappet face stress versus maximum spring load for various cam nose radii



■ Fig. 10 – Comparative tappet face stresses for 100-in. spherical, 30-in. spherical, and flat faces – based on 0.100-in. cam nose radius



■ Fig. 11 – Tappet face stress versus engine rpm – 30-in. spherical face cast-iron tappet on steel cam

0.100-in. nose radius, for the various types of faces. The ordinate represents the spring load, and the abscissa the corresponding stress. The solid lines show the stress for each type of face with this particular nose radius. The dotted lines represent the difference. This chart alone would indicate that a flat face would be best in any case but, as previously shown, misalignment can affect the stress of the flat face to such an extent that 125,000 psi theoretical stress must be accepted as the maximum allowable. The spherical face operates successfully up to 189,000 psi maximum stress. Therefore, the stress on a flat-face follower would have to be less than on a spherical face by 64,000 psi in order to show better results. This indicates that, where the cam nose is a 0.100-in. radius, the 100-in. spherical face would be best in service because the greatest theoretical gain of the flat over the 100-in. radius is 50,000 psi. The 30-in. spherical radius face would not show as good results as the flat face because the flat face is theoretically more than 64,000 psi stress better.

Although the stress is greatest when figured statically, we know that the rubbing velocity enters into the results. The picture with respect to the stress velocity curve would look somewhat like the chart shown in Fig. 11 and, while calculating the stress statically is not theoretically correct, one can see from this chart that it is practical and can be used as a basis of comparison, which conclusion field results have borne out.

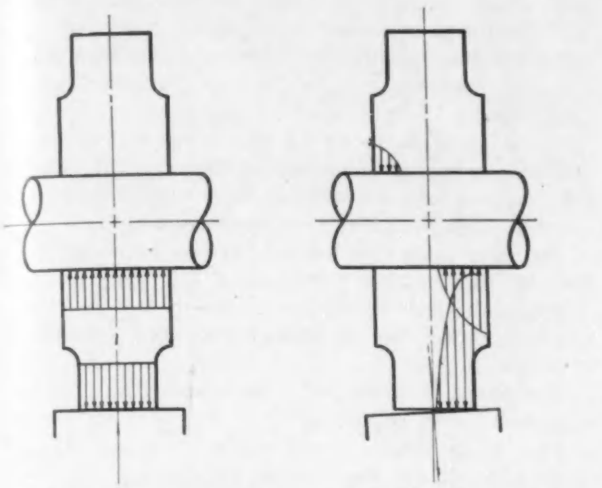
If the curve representing stress crosses the stress-velocity curve, failure will occur at speeds anywhere between the two points of intersection. The maximum stress, which has been determined through experience, establishes a stress curve which does not cross the stress-velocity curve.

While no field information on the roller-type follower is at present available, it is obvious that the same general conditions of stresses would occur. In the case of misalignment between a flat-faced roller and cam, as indicated in Fig. 12, false stresses would exist such as are shown for the flat-faced tappet. In addition, the bearing between the roller and the pin could be overstressed, and it is very likely that this excessive stress causes much of the trouble with seizure between the roller and the pin. A crown roller, as shown in Fig. 13, would relieve this condition, and has already been put in production on some aircraft engines with excellent results. The amount of crown would depend upon the clearances and deflections. Where this roller is in production, it is around a 5 to 6 in. radius.

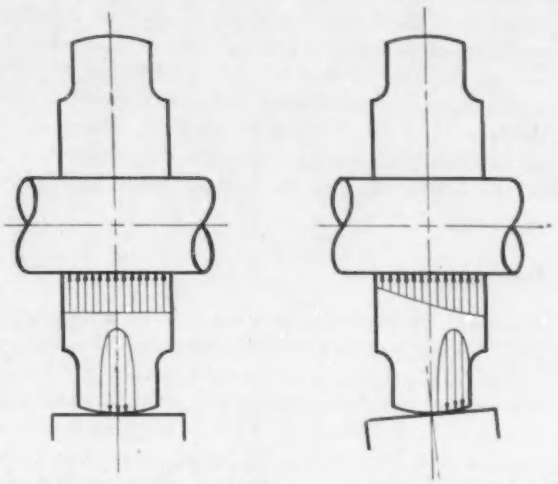
It appears that, if the consideration is chiefly the prevention of scoring between the roller and the pin, a smaller radius would be desirable as this design would keep the load distributed over the length of the pin with a greater amount of deflection. However, if the chief interest is the contact between the face of the roller and the cam, then a much larger radius would be desirable as long as there was a sufficient amount to take care of the misalignment. Therefore, it would probably be advisable to make tests to determine the best radius for any given case depending upon the type of failure involved.

The following formula for calculating the stresses between a cam and roller has been derived from Timoshenko's "Strength of Materials":

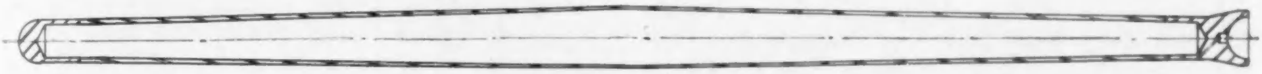
$$S_{max.} = 0.59\sqrt{\frac{P(R_1 + R_2)}{W \cdot R_1 \cdot R_2\left(\frac{1}{E_1} + \frac{1}{E_2}\right)}};$$



■ Fig. 12 – Stresses between straight cylindrical roller and cam – without deflection and with deflection



■ Fig. 13 – Stresses between crowned cylindrical roller and cam – without deflection and with deflection



■ Fig. 14 – Optimum double-tapered pushrod design

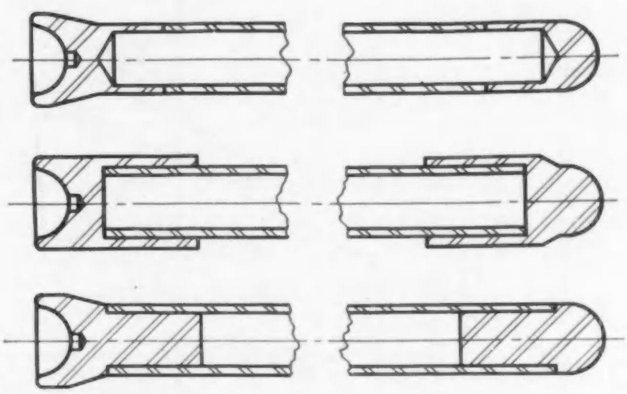
Where: P = the load on the cam
W = the cam width
R_1 = roller radius
R_2 = radius of curvature at cam
E_1 & E_2 = the modulii of elasticity of the materials in contact

In the case of the roller follower and the tangential cam with a given timing, accelerations and decelerations are a rather cut-and-dried proposition unless we resort to the expensive means of providing a hollow-flank type of cam by grinding with a small-diameter wheel. The deceleration is relatively high and the acceleration is relatively low; whereas, in the case of the mushroom cam, the deceleration is relatively low and the acceleration is higher. This condition means that, for a tangential-type cam and roller follower, the higher load-carrying ability of the roller is partially offset by the higher requirements resulting from higher deceleration and heavier parts entailing heavier spring loads. Therefore, in many cases where roller followers are used, mushroom followers can be substituted successfully to advantage both from performance and cost viewpoints. This condition is not always true, but there are many engines in which a study of the possible mushroom cam design and the loads involved as compared to the tangential design, would indicate the successful use of mushroom followers.

It is apparent from the foregoing that camshaft deflection should be held to a minimum from the standpoint of both false motion of the valve gear and stresses between the cam and follower. It is recommended that shaft deflection be held to 0.002 to 0.003 in. under maximum load. In order to attain this degree of rigidity, it may be desirable to use a hollow shaft unless weight considerations are of no importance.

■ Pushrods

A theoretically correct pushrod would be of a double-tapered design, an approach to the optimum shape being illustrated in Fig. 14. This, however, is a point which is not usually considered of sufficient importance to warrant the extra work involved in production of the pushrod. So, returning a little more to the practical side, the push-



■ Fig. 15 – Commercial pushrod designs

rod should at least be of a tubular design with as large a diameter as available space will allow and with a sufficient cross-section of material to provide maximum rigidity with minimum weight. Probably the lightest design of this type of pushrod is the welded end construction as shown at the top, Fig. 15. The advantages over many conventional designs with respect to weight are apparent. The other views show some of the types of end construction in common use.

■ Rocker Arms

Obviously, the rocker arm should be designed for the greatest rigidity with a minimum of weight. Since little side loading is imposed on the rocker arm, the greatest section of material should be in the plane at right angle to the axis of the bearing. Heavy sections in the extremities should be avoided. In some cases, considerable weight may be removed from rocker arms by making a study of the stress pattern in plastic models, which is the subject of several engineering papers.

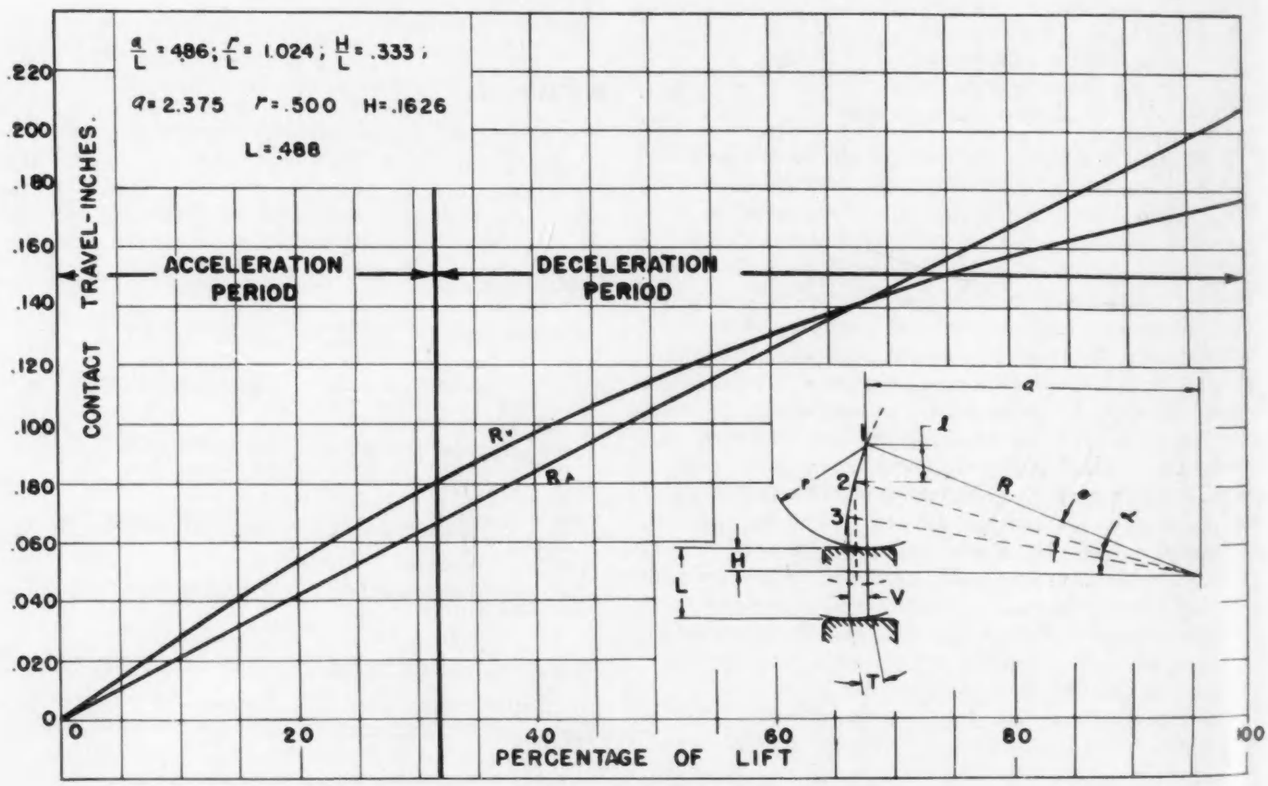
Rocker-arm ratios in common use vary from 1:1 to as high as 2.5:1. Either of these extremes would be detrimental to valve-gear performance. In the case of 1:1 ratio, the pushrod and tappet are moved as rapidly as the valve, and the total inertia of the valve gear is unnecessarily high. The cam design would then parallel that of an L-head engine with the disadvantage of the additional reciprocating weight.

In the opposite extreme, excessive deflections enter into the rocker arm due to the additional lever advantage. In addition, deflections of the cam and other parts on the cam side of the valve gear are multiplied by the rocker ratio and, where small actual deflections exist, this deflection can amount to a considerable figure at the valve. Cam errors are multiplied at the valve, and clearance changes resulting from the camshaft side of the valve gear are exaggerated.

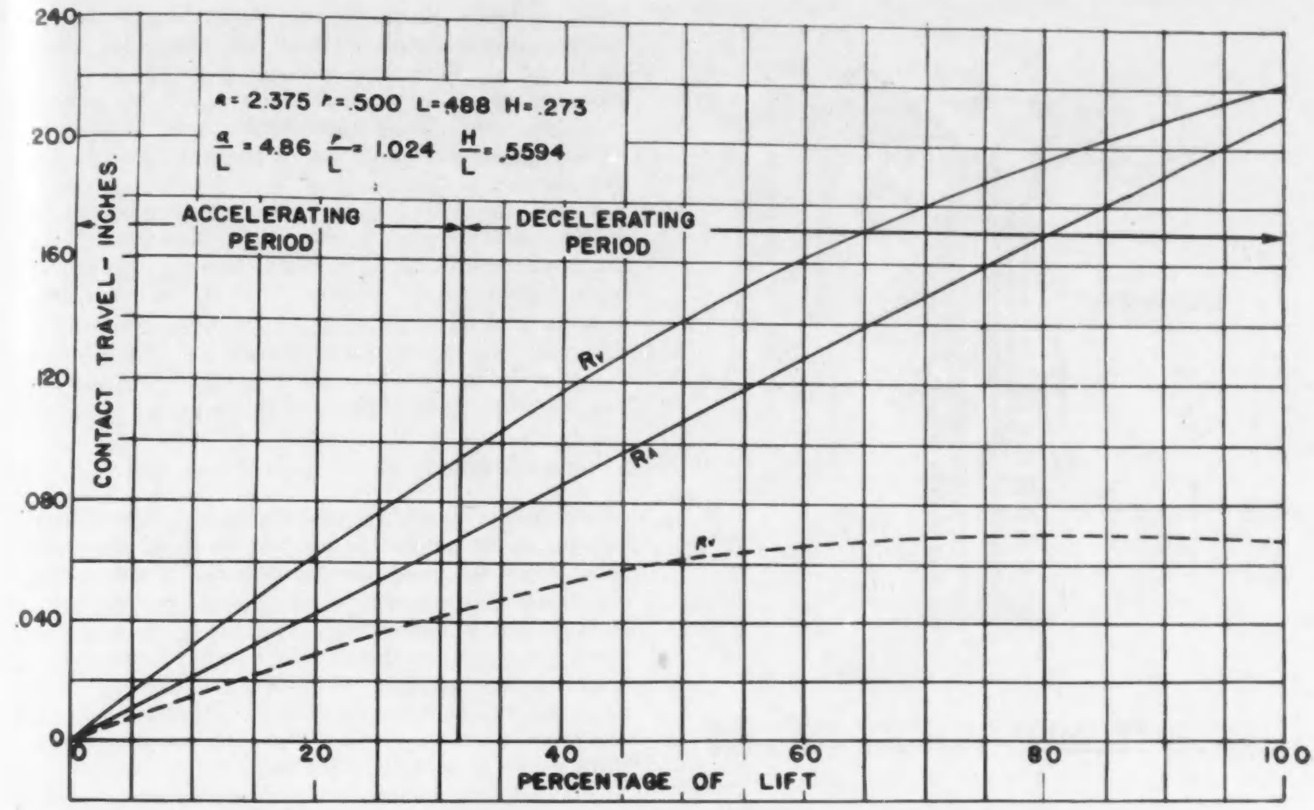
Following are some of the factors to be considered in determining rocker ratio: body diameter of the shaft and the cam lobe height which can be assembled through the bearings; deflections under maximum loading; total inertia of the valve gear; unit loading between parts; and clearance changes between parts.

When all these conditions are taken into account, a moderate rocker ratio of about 1.3 to 1.6:1 probably will be selected.

The geometry of the rocker arm, with respect to valve movement, is an important item. It is a rather common practice to divide the travel of the valve half and half above and below a line through the center of the rocker arm normal to the valve stem. This arrangement reduces the total slippage of the contact point on the valve stem to a minimum, but does not provide the best possible condition. The movement should be related to load in such a way that the greatest amount of slippage occurs under the lightest loads, and minimum slippage takes place during acceleration. Such an arrangement will reduce the side load on the valve and tend to prevent excessive wear of the valve stem in the guide. Thus, in the case of mushroom cams, in order to provide minimum slippage during acceleration and maximum during deceleration, approximately one-third of the valve lift should occur above and two-thirds below the centerline. This general



■ Fig. 16 – Chart for satisfactory rocker geometry for mushroom cam



■ Fig. 17 – Chart for unsatisfactory rocker geometry

principle applies to roller and elephant's-foot types as well as the ordinary shoe contact.

Fig. 16 shows a favorable rocker-arm geometry for a mushroom cam. The margin between the two curves represents the slippage that takes place during valve opening or closing. Note that, with this ratio, namely 0.333, the greatest amount of slippage would occur from the point where the lines cross, about 67% opening to 100% opening and back to 67%. In other words, this slippage would occur during the deceleration period.

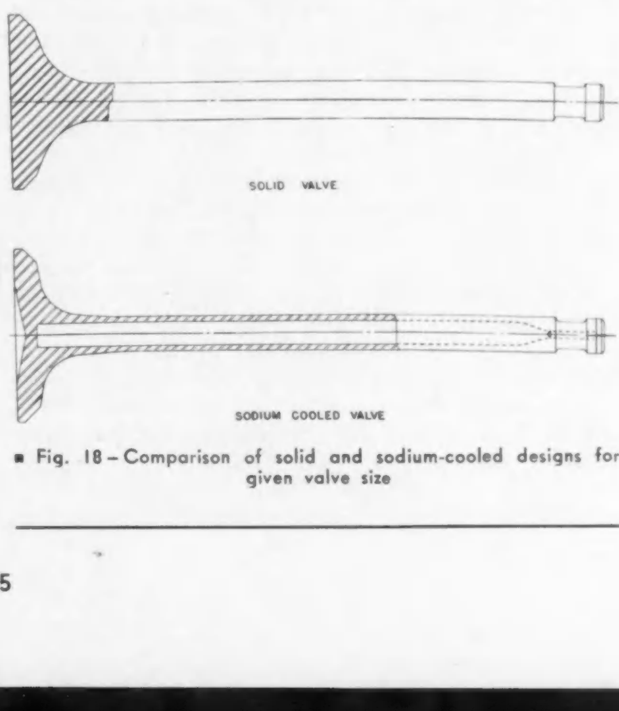
Using the same camshaft, a ratio of 0.5594, as shown in Fig. 17, would provide less slippage during deceleration and more slippage during acceleration. Although this would provide a lower total slippage, it would create great wear due to the fact that maximum slippage occurs under heavy loading.

Besides establishing a favorable rocker-arm geometry to reduce side loads on the valve, the use of an elephant's foot or its equivalent is highly desirable in large or high-output engines.

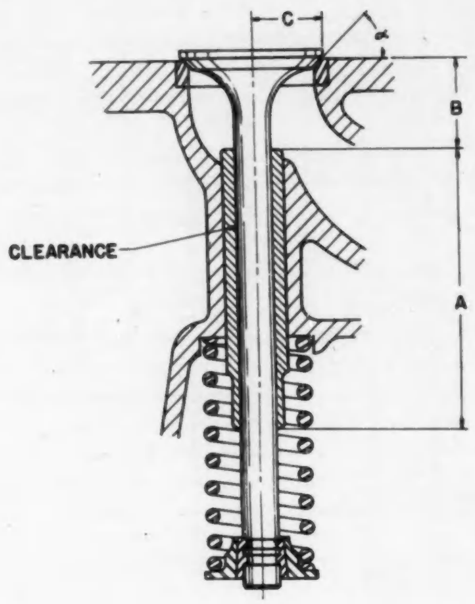
■ Valves

The valve should be of a material having a high hot strength and stainless properties, and the foot end should be hard. It is not the purpose of this paper, however, to go into any detail on materials. The solid valve has made it necessary to keep the stem diameter down in the interest of lightness, in some cases to a point where the valve stem is inadequate as a bearing.

Sodium-cooled valves have received little consideration for diesel engines, mainly because additional cooling has not been required. However, with continually increasing speeds, it becomes necessary to reduce the weight of the reciprocating parts of the valve gear. One point where considerable reduction in weight can be made is in the valve itself. When the valve stem is drilled, there is a loss in cooling capacity, particularly with respect to the heat carried down the stem of the valve. This loss can be considerably more than offset by partially filling the valve stem with sodium. Taking, for example, valve size *X* (see Fig. 18), material could be removed as shown



■ Fig. 18 – Comparison of solid and sodium-cooled designs for given valve size



$$\text{RAMP} = \frac{(.5\,A+B)\ \text{TAN}\,\alpha + C}{A} \times \text{CLEARANCE} + \text{DEFLECTION}$$

■ Fig. 19 – Formula for determining closing ramp requirements

in the lower view which would reduce this valve to 76% of its original weight. Referring this weight reduction to the chart indicating the accelerating load, it can be demonstrated that, by removing this amount of weight from the valve, it will be possible to run the valve gear about 4% faster with a given accelerating load.

■ Valve Guides

Valve guides may be of any material that will run successfully with the valve. The fit in the head should have as good a thermal contact as possible with a smooth finish in both the hole and on the outside diameter of the guide. It should have as tight a press fit as can be used without causing breakage, distortion, or scoring in pressing the guides in. The heads should be designed to provide circulation of the coolant as close to the guide as possible.

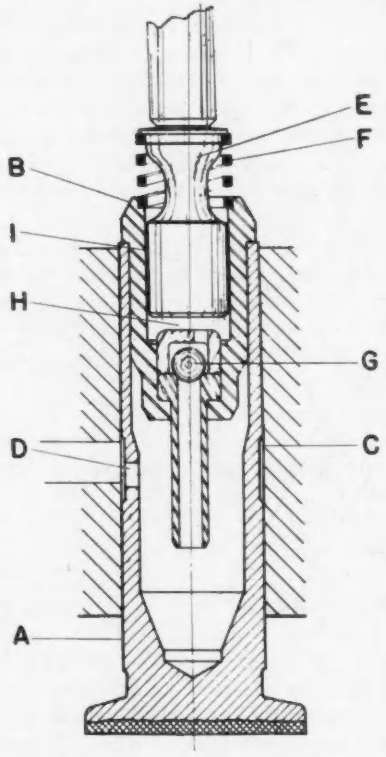
It is obvious that, as soon as the valve is lifted from the seat, it will assume an angularity with respect to the axis of the guide to the greatest extent that clearance of the valve stem in the guide will permit. The valve will move up in this cocked position and return in the same position, sliding over to line up with the seat after contacting the seat on one side. Thus, if the valve contacts the seat at a point high enough on the flank of the cam, velocity can become such as to cause a dishing of the valve by means of a peening action. A method has been evolved, which is used in designing cams, for determining the closing ramp requirements. The formula is shown in Fig. 19. Included in the total ramp requirements are either the actual or estimated deflections, and the tappet clearance where hydraulic lifters are not used. As this formula indicates, it is important to extend the guide as far toward the head of the valve as possible, and to use the minimum allowable valve stem to guide clearance. If this practice is followed, the closing ramp need not be excessive to prevent the valve from contacting the seat at too high a velocity.

In the case of the exhaust guide, the boss should be extended with the guide in the interest of cooling. There is a trend toward making guides to finished sizes so that better finishes can be provided, than by the more conventional method of pressing the guide in and reaming it in place. Guides of permanent-mold cast iron, finished to size and Ferroxed, have proved very successful in at least one automotive engine application.

■ Valve Springs

Valve springs should, generally speaking, have as high a frequency as possible without overstressing the steel. There are many cases where breakage occurs with a very low theoretical stress because spring surge causes stresses far in excess of the theoretical. Engineers with the Eaton Spring Division report that they have corrected many cases of valve-spring breakage by increasing the frequency of the spring, which necessarily raises the stress.

In this connection, it may be pointed out that spring length should be held to the minimum necessary to obtain the required loads without overstressing the material. This purpose can be accomplished by raising the spring seat or



■ Fig. 20 – Automotive application of hydraulic valve lifter

by moving the retainer farther from the valve tip. If neither of these steps is feasible, it may be necessary to resort to the use of spacers.

In designing valve springs, it is usually found that the critical factor with regard to load is the requirement at the point of reversal. The best practice is to provide a load safety factor of 30% at this point to allow for momentary reductions in load owing to surge.

■ Seat Inserts

Valve seat inserts should be used if for no other reason than to provide a replaceable seat. Materials range from plain cast iron, to be used where the replaceable feature is the chief consideration, to stellite-faced inserts for high-output operation. The particular material and design selected, of course, depend on the design and characteristics of the engine.

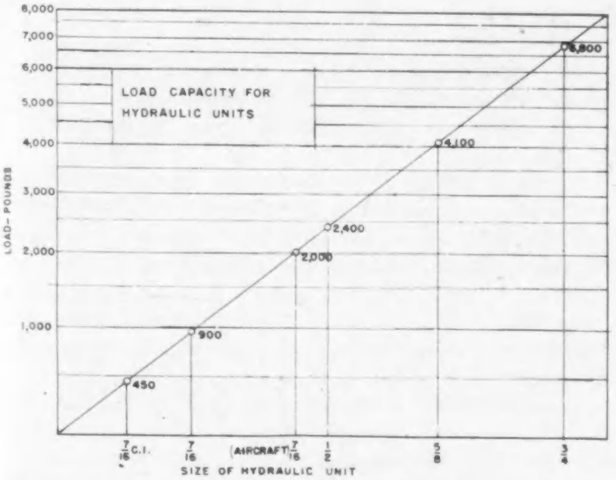
Inserts should be installed with the minimum possible interference, small to medium-sized rings with 0.002 to 0.004 in. and larger rings with 0.003 to 0.005 in. Ample depth tends to prevent loosening of the ring. A smooth finish on counterbore and ring is necessary for thermal contact. However, the most important factor in the design of seat inserts is undoubtedly resistance to distortion, and insert sections should be designed for the maximum possible rigidity.

■ Hydraulic Lifters

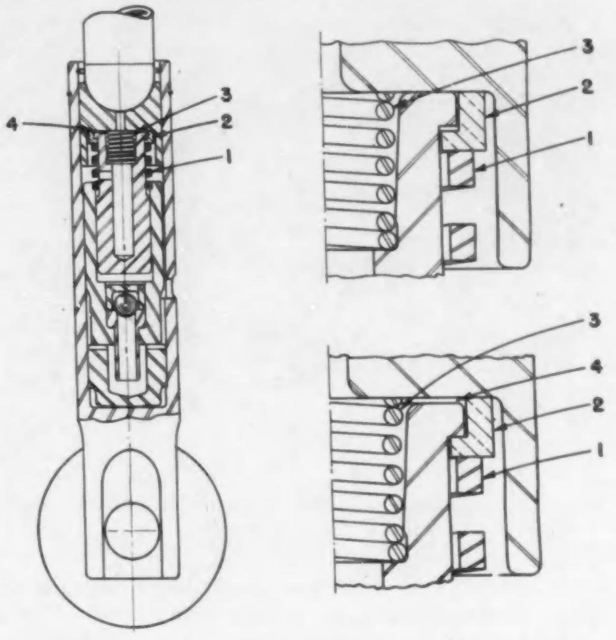
Hydraulic valve lifters are used in a considerable number of automotive, aircraft, and diesel engines. Fig. 20 shows an ordinary automotive application and will be used to describe the operation of the lifter.

The assembly is composed of the lifter body, *A*, which corresponds to the ordinary tappet, and the hydraulic unit or compensating member, *B*. The unit is removable from the body and is essentially universal for automotive installations.

Oil is supplied to the lifters at regular engine pressures. An annular groove, *C*, is provided on the tappet body to maintain registration with the oil feed, and the oil is delivered to the lifter body through the drilled opening, *D*, in line with the annular groove.



■ **Fig. 22 – Load capacity for different sizes of hydraulic units by plunger diameter**



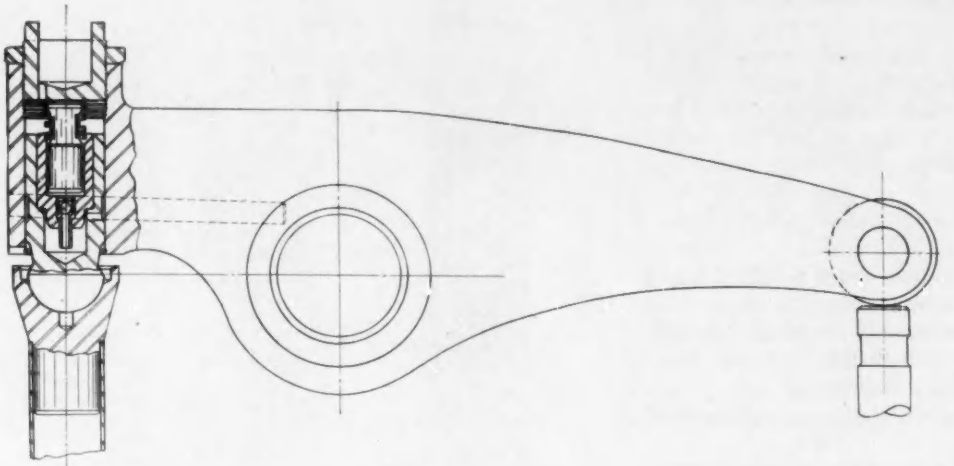
■ **Fig. 21 – Eccentricity compensator – radial-engine installation**

In a complete cycle of the cam, starting with the lifter on the base circle, the hydraulic plunger, *E*, is actuated outwardly by the plunger spring, *F*, to take up any clearance between the end of the valve stem and the cam. As the cam revolves, the initial pressure developed seats the check ball, *G*, so that oil under the plunger is trapped and the valve is lifted on a column of oil at *H*. During the interval when the valve is lifted off its seat, a slight oil leakage occurs between the plunger and cylinder at *I* which is necessary to compensate for any expansion in the valve gear. During the valve-closed period, the chamber below the plunger is replenished with oil, thereby eliminating all clearance.

In some cases, such as radial aircraft engines where the runout of the cam is considerable, the valve may be cracked open for a part of the base-circle travel. In order to overcome this, an "eccentricity compensator" can be incorporated in the unit. While different types of eccentricity compensators have been developed, the principal one is shown in Fig. 21, a radial engine installation. During the time the unit is filling, the horseshoe collar (2) provides a stop for the spring (1), and the plunger is held down against the shoulder in the washer by the small spring (3). This arrangement provides a pre-determined amount of clearance at (4). This amount of movement will then be required before the valve can be lifted.

The various sizes of units are rated according to the plunger diameter (see Fig. 22). The size selected for a given engine depends on valve gear loading, most large diesel engines requiring the ⅝-in. unit, which will withstand a maximum load of 4100 lb.

The hydraulic unit may be located in the tappet, in the



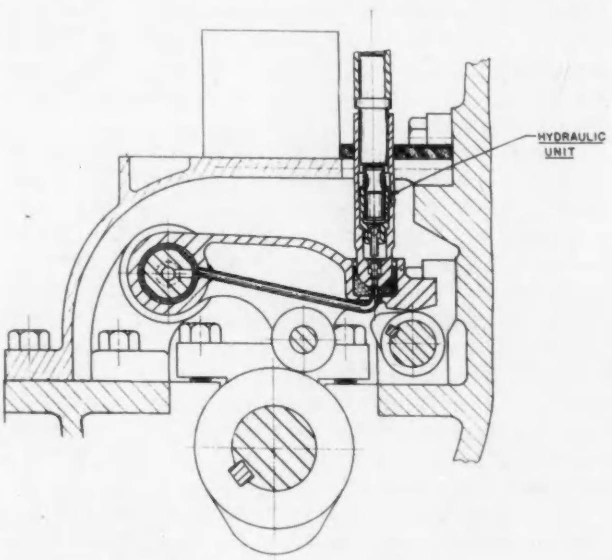
■ Fig. 23 – Hydraulic valve lifter applied to large diesel engine

upper or lower end of the pushrod, or in the rocker arm. When installed in the rocker arm, the unit is usually placed on the pushrod side but, in a few cases, it has been applied to the valve side. In addition to these applications, in at least one experimental case, the unit was located at the fulcrum point of the rocker arm.

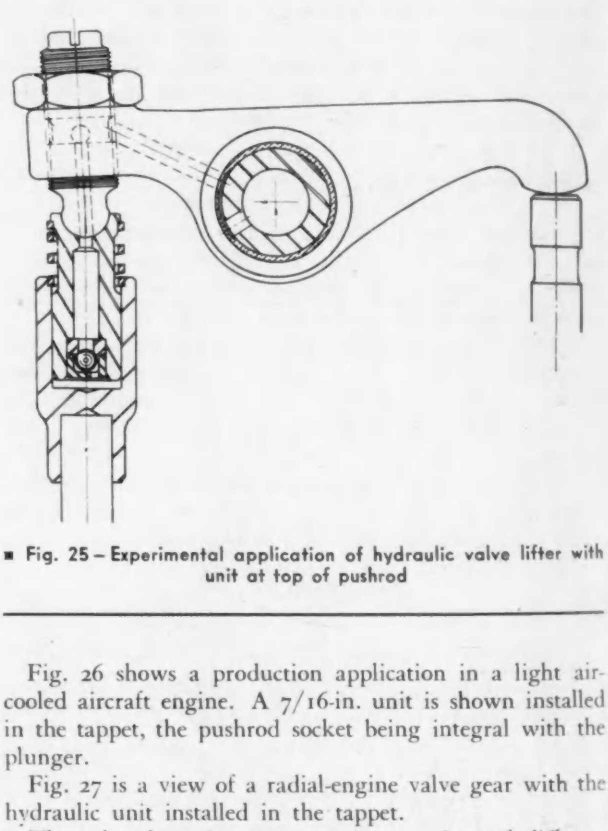
The application in the most common use in diesel engines, is the rocker-arm type with the unit installed on the pushrod end. Fig. 23 shows the hydraulic lifter as applied to a large diesel engine.

Fig. 24 shows an unusual application where a ½-in. unit is applied to the lower end of the pushrod.

Fig. 25 shows a type of application with the unit at the top of the pushrod, which has been tried experimentally. While it has operated successfully in several engines, there has been no occasion to use the application in production.



■ Fig. 24 – Unusual application of hydraulic valve lifter – ½-in. unit applied to lower end of pushrod



■ Fig. 25 – Experimental application of hydraulic valve lifter with unit at top of pushrod

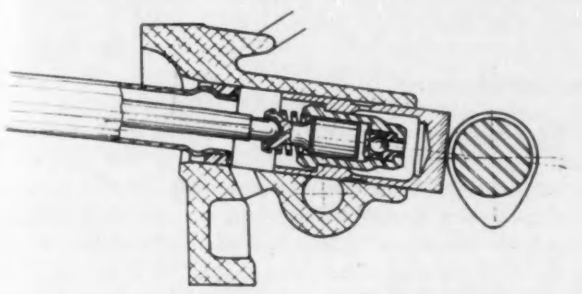
Fig. 26 shows a production application in a light air-cooled aircraft engine. A 7/16-in. unit is shown installed in the tappet, the pushrod socket being integral with the plunger.

Fig. 27 is a view of a radial-engine valve gear with the hydraulic unit installed in the tappet.

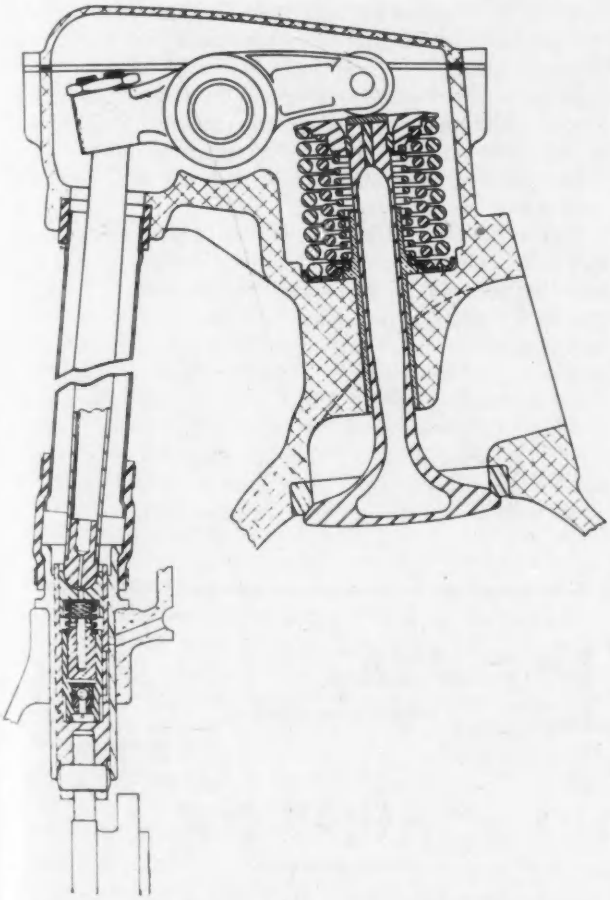
There has been in common use a variety of different means for axially shifting the camshaft in a direct reversible engine. The Wilcox-Rich Engineering Department, working in conjunction with Baldwin-Locomotive, has developed a method of lifting the tappets for shifting the camshaft, which may be of some interest.

This design was applied experimentally to the Baldwin VO engine, which is a 12¾-in. bore, 15½-in. stroke, 625 rpm, 8-cyl model (see Fig. 28). When there is no oil pressure to the tappet body, the spring, *A*, at the top of the tappet holds the tappet off of the camshaft to the extent of the travel of the hydraulic unit which, in this particular case, is ⅞ in. When the oil pressure is applied.

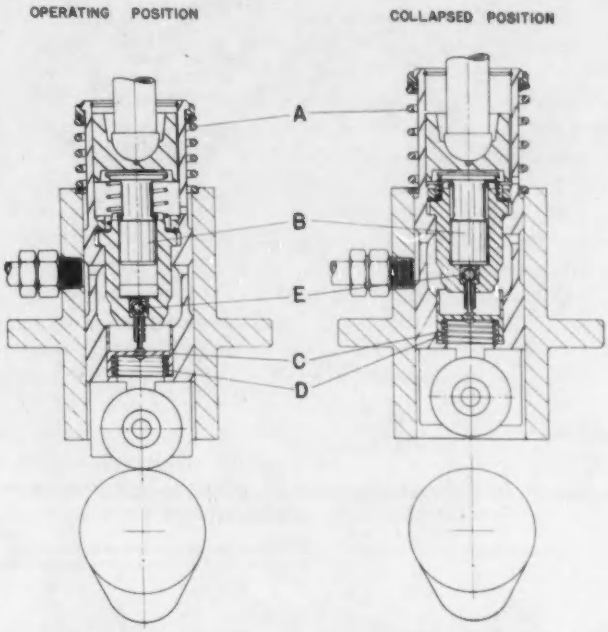
the pressure acting on the area of the plunger, B, overcomes the body spring, A, and forces the tappet down on the cam, at the same time supplying the oil for hydraulic operation of the lifter. The oil pressure also exerts a force on the lower piston, C, which overcomes the spring, D, in the bottom of the bore. When the oil pressure is shut off, this piston moves upward, actuated by the spring, lifting the check ball, E, off of the seat. At the same time, the body spring moves the assembly upward, causing the



▪ **Fig. 26 – Production application of hydraulic valve lifter in light air cooled aircraft engine**



▪ **Fig. 27 – Hydraulic valve lifter application for radial aircraft engines**



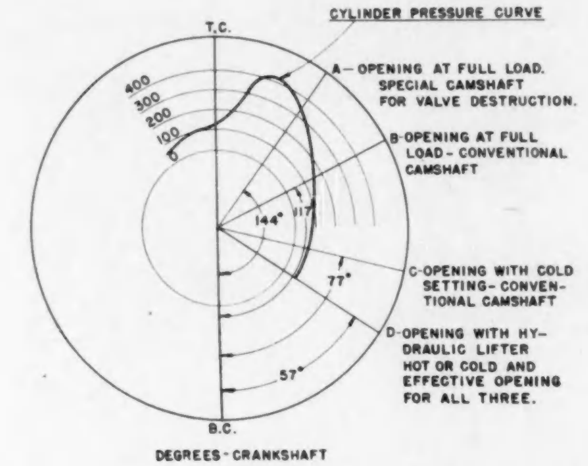
▪ **Fig. 28 – Method of lifting the tappets for shifting the camshaft for reverse operation – Baldwin VO 8-cyl engine**

oil to leak past the check valve out of the high-compression chamber of the hydraulic unit and the roller to clear the base circle of the cam by ⅞ in. This permits shifting of the cam for reverse operation.

Tests indicate that with SAE-40 oil at 135 F and 125 psi pressure, the assembly requires 0.4 sec to expand to the operating position, shown at the left, and 0.3 sec to collapse into the shifting position, at the right. Using the same oil at 32 F, 1.6 sec are required to expand and 3.0 sec to collapse.

The advantages of the use of hydraulic lifters are: increased valve gear life, protection against broken parts, freedom from adjustment, increased flexibility in cam design, and quietness.

Where hydraulic lifters are used, the timing remains constant throughout the entire temperature range of the engine, and is unaffected by wear of any of the reciprocating parts. This is a very great advantage in most aircooled engines or any engine in which tappet clearance increases as the temperature rises, as it obviates the necessity of compromise in the cam design for starting and idling. It is also decidedly advantageous in any engine in that it eliminates the necessity for using a long ramp in order to lift and seat the valve at low velocity throughout the temperature range. The use of long ramps imposes an extra amount of punishment on the valve, example of which is shown in Fig. 29. This chart was taken from data obtained on an ordinary automotive engine, and represents a radial timing diagram upon which is superimposed the cylinder pressure curve in heavy line. The manufacturer gives D as the timing point for the opening of the exhaust valve, but allowable clearance variations bring the cracking point of the exhaust valve to C. At full throttle, maximum power, the clearance is reduced to



■ Fig. 29 – Radial timing diagram of automotive engine on which is superimposed the cylinder pressure curve

bring the cracking point of the exhaust valve to *B*. In order to accelerate valve destruction tests, for which this engine is used, a camshaft was designed with a special ramp which brought the cracking point of the exhaust valve to *A*. This arrangement cut the test time about in half; in other words, moving the cracking point from *B* to *A* doubled the destruction of the valve.

Fig. 30 shows three ramps; the center one is the conventional ramp used on this engine in production; the upper one is the special ramp for valve destruction; and the lower one, or rather the lift curve shown without a ramp, is the one which may be used with hydraulic lifters.

In working with engines in which the tappet clearance increases as the temperature rises, cases have been found where the tappet clearance was adjusted hot, and the valves were damaged in starting cold before reaching normal clearance. Likewise, with clearances adjusted cold, they opened up a sufficient amount when the engine reached running temperature to move the closing point away from the ramp of the cam and close the valve at excessive velocity, causing breakage. Hydraulic lifters in such engines have corrected both these types of trouble.

In considering engines in which valve clearance tends to decrease as temperature rises, one of the common problems is to allow sufficient clearance to take care of extreme operating conditions. It is obvious, since a considerable percentage of the cooling of a valve is accomplished by transferring the heat from the seat of the valve to the block, that, when the valve is held open by some temporary cause, the temperature of the valve rises. Because of the resulting expansion in the valve itself, this rise in temperature often may be sufficient to cause the valve to remain in the open position even when the initial cause for holding the valve open is removed. Thus, it may not seat again until the load has been reduced to permit sufficient cooling.

The use of hydraulic valve lifters is an assurance that valves cannot hold open under any operating conditions except when this condition is caused by stem deposits or broken springs. Even then, when the initial cause is removed, the valve will close quickly and, if too much damage has not already been done, will continue to give satisfactory service.

An initial adjustment is all that is necessary for hydraulic lifters, and this adjustment is unnecessary when the parts can be held to limits that will put the units within their operating range. This range is usually ⅛ in. but, in some cases, has been made as much as 15/16 in. and as little as 1/16 in., depending upon the individual requirements. This adjustment need not be made accurately and can be done by giving the adjusting screw a certain number of turns from either extremity of the range of the unit. After this initial adjustment, compensation for change in clearance is assured.

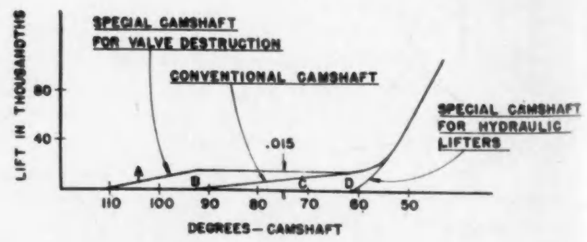
■ Conclusion

To summarize briefly the various points set forth in this paper:

Considering all factors, it is recommended that the camshaft deflection be held to not more than 0.002 or 0.003 in. under its maximum load. In order to accomplish this purpose, it may, in some cases, be desirable to use a hollow shaft. Accelerations should be as high as possible on both the opening and closing side, and the opening side may range up to three times that of the closing side. Cam followers should not be overstressed by the use of too small a nose radius. If mushroom-type followers are used, they may have a cast-iron face when operating on cams of any material. Steel followers can be used successfully only on cast-iron cams provided that the cam is properly designed for best performance. The latter combination is sometimes advantageous from a reciprocating-weight standpoint, but the section of the cast-iron cam should be about 30% greater for equal rigidity.

The pushrod should be tubular and may well be of the welded-end construction.

The rocker arm should be of the lightest construction that will not deflect excessively under maximum load and heavy sections should be avoided near the ends. A rocker ratio of 1.3 to 1.6 is best when stresses, speed of the valve gear, and other factors are considered. It is usually advisable to use some sort of elephant's foot to avoid concentrated loading on the valve stem end and its accompanying side pressures on the valve. The use of a roller is very good practice in large engines where loads are high. Whether this is used or not, the geometry of the rocker arm should be such that the movement of the valve



■ Fig. 30 – Ramp for conventional camshaft, special ramp for valve destruction, and lift curve with special camshaft for hydraulic lifters

is about one-third above and two-thirds below the centerline. The ratio will vary between a tangential cam and a mushroom cam, but the geometry can be plotted to determine the best condition for a given cam.

The valve should have a hard tip and be constructed of a material having a high hot strength and stainless properties. Where relatively high speeds are required, a hollow drilled valve is desirable for lightness. If this construction causes excessive heating, sodium cooling should be used in either exhaust or intake. The head should be so shaped as to resist dishing with as little mass as possible and, in some instances, the hollow head may be used to advantage.

Valve stem guides may be of any material that will run successfully with the valve, but should be extended as high as possible on the valve to permit ample clearance without excessive valve cocking. The valve must not be permitted to strike the seat until it is slowed to ramp velocity. In the case of the exhaust guide, the boss should be extended with the guide in the interest of cooling.

The spring should have a safety margin of at least 30% at the point of reversal and should have the highest possible frequency without overstressing the material. Spring breakage often occurs in cases where the theoretical stress is very low because of surging which sets up stresses far in excess of the theoretical.

The valve seat insert should be of ample section, and of a material which provides as much resistance to distortion as possible. It is important that it should have maximum rigidity.

The most important advantages of hydraulic lifters are increased valve gear life; elimination of frequent adjustment and down time; consistent engine performance in so far as the valve gear is concerned; and the possibility of increased engine performance through changes in cam design. These and other advantages attending the use of hydraulic lifters make them a requisite in any modern engine design.

This paper is a product of the personnel of the Wilcox-Rich Laboratory. The mathematical work of M. C. Turkish and the literary and engineering work of J. O. Eaton, Jr., are especially acknowledged.

Lightplane Design and Development

THE lightplane, designed primarily to provide inexpensive private flying, has imperceptibly drifted into the military training program. This support by the government has made possible a rapid increase of production facilities for lightplanes that would not be justified for civilian needs alone. Regardless of comments to the contrary, the lightplane has demonstrated its suitability for the preliminary flight training and selection of military pilots.

However, on the civilian side of the picture, the phase of flying for which the lightplane was supposedly developed, very little has been done. In this case the training of the pilot is not an end in itself, but only a means to an end. In fact, too many people in the business seem to think that the primary use for an airplane is to teach people to fly. This is understandable, since flying instruction has been the only sure source of revenue in the history of the business. As a result of this trend, too many pilots have quit flying after the thrill and novelty has worn off and too many private owners have sold their airplanes for the same reasons after a short period of ownership.

The lightplane industry has put flying within the reach of many people. Unfortunately, however, flying by itself is not enough to provide support to maintain the industry after the present emergency, and the industry is going to stand or fall according to its ability to produce transportation rather than recreation. While it is true that many factors affecting the usefulness of the lightplane are outside of the control of the industry, the fact remains that the overall performance of the airplane is bound to be the chief contributing factor.

The methods of making the lightplane more useful should then be the guides in its development. Many suggestions have been made to make flying more useful. In general, these can be classified as attempts to increase the high speed or to improve the take-off and landing performance.

The quest for higher cruising speeds has been a continuing one in the development of aeronautics. However, since low landing speeds are incompatible with high cruising speeds, it appears likely that fast airplanes will continue to be "airport" airplanes for some time to come. Moreover, the factors that are most effective in increasing the speed of an airplane also rapidly increase its cost. In addition, the design changes that tend to increase the speed of an airplane also tend to increase the skill required to fly it.

Whether development should emphasize the high or low speed characteristics of the lightplane depends upon the demands of the user.

In my opinion, the low-speed type will be the more useful to the average man. Automobile experience shows that the average trip is less than 9 miles. If for some reason the automobile were not practical for trips of less than 50 miles, it is doubtful that very many would be used. The average lightplane is not useful for trips of less than 150 miles; for shorter trips it is more convenient to go by automobile. Very few of us have the occasion to travel that far very often and consequently the airplane is not very useful to us. However, if the airplane could be made practical for trips over 15 miles it would then become very useful.

■ Landing Speed

Since the minimum speed possible is proportional to the square root of the weight divided by the product of wing area and maximum lift coefficient, it is obvious that development should be in the direction of increasing the maximum lift coefficient. Although it is possible to obtain the required low landing speed by the use of low wing loadings, it is not desirable to do so from the standpoints of comfort at high speed, handling in winds, and wing structural weight. For instance, the Wright biplane could be flown comfortably at 25 mph because of its 2 lb per sq ft wing loading. Even with modern wing sections, this per-

formance could not be very much improved.

However, it is obvious that if high lift is required, it should be obtained by increasing the maximum lift coefficient. For instance, if a slotted flap is used in conjunction with a leading edge slot, a lift coefficient of double that of a good wing section can be obtained. This allows halving the wing area for the same landing speed. With this wing arrangement, an honest 30 mph landing speed can be obtained with a wing loading of about 7 lb per sq ft.

If for simplicity of construction the front slot is fixed to be open at all speeds, the wing drag will be about equal to that of a plain wing giving the same maximum lift coefficient. For the same wing aspect ratio, the plain wing would provide better initial climbing performance because its span would be 40% greater than that of the slotted and flapped wing and its profile drag would be lower.

With a fixed wing airplane it becomes difficult to obtain landing speeds of less than 30 mph. However, since wind speeds of 10 mph are quite common, the actual ground speed of an average landing would be nearer 20 mph. There appears to be very little to be gained by further reducing the landing speed. It should be noted that these speeds are actual and not "sales-department" speeds.

That an airplane requires a much smaller space for landing than for take-off is common experience. The angle of descent in a glide and the ground deceleration with brakes, have in general been greater than the ground acceleration and angle of climb after take-off. Except for safety in forced landings, there is no point in being able to get into smaller areas than you can get out of.

Take-off Run

As is well known, the length of ground run is a function of the acceleration and the take-off speed. The minimum take-off speed for most airplanes is lower than the landing speed, chiefly because of the slipstream effect on the wings. This is particularly true when flaps are used to shorten the take-off run. The take-off run can be expressed as follows:

$$S = \frac{V_t^2 W}{64 T_e}$$

where S is the take-off run in ft
V_t is the speed at take-off in fps
T_e is the net thrust at a speed of $0.7V_t$
W is the airplane weight in lb.

Obviously for a given take-off speed the ratio of weight to net thrust should be as low as possible. Increases of thrust can, of course, always be obtained by increasing engine horsepower. However, since the propulsive efficiency for this type of airplane at the speed appropriate for the take-off run is about 30%, increases of engine power are not very effective in increasing the thrust.

Important gains, however, can be made by increasing propulsive efficiency. The steps that might be taken to increase the thrust at low speeds are:

(1) Gearing to reduce slipstream losses by increasing propeller diameter.

(2) Controllable pitch propellers.

(3) Gearing and controllable pitch propellers.

(4) Two-speed transmissions and controllable pitch propellers.

(5) Reduction of blade angle by increasing blade area.

Climb

The angle of climb after take-off is as important as the ground run in the ability to get out of small fields. Moreover, the speed at which a steep angle of climb occurs should be close to the take-off speed, so that little distance is lost in accelerating from the take-off speed to the climbing speed.

The high thrust available from the large diameter, variable pitch propeller in the take-off condition will also provide the high thrust necessary for a high angle of climb at the take-off speed. The additional airplane design quantity that is important for this condition is the span loading, W/b^2, which should be kept as low as possible. When high lift devices are used to decrease wing area, the designer's tendency is to think in terms of wing aspect ratio rather than span, and consequently take-off and climb performance is likely to suffer when the design includes such devices. An angle of climb of 15 degrees should easily be possible with the large propeller and overspeeded engine.

Noise

The additional weight required to produce propeller design characteristics that lead to optimum take-off and climb performance may possibly be justified on the basis of reduced noise level alone. One of the chief factors leading to fatigue in flying is noise, and there can be no question but that flying would also be more pleasant if the noise level were reduced. Because of the light-weight windshields and windows used, and because of inevitable cracks around lightly built doors, it is unlikely that important reductions of sound level can be made by the soundproofing methods used so effectively in transport airplanes. A direct reduction of noise at its source appears to be a more promising method in the case of the lightplane. Engine mufflers have already been developed to a high state of efficiency, but the improvement to be expected in this direction alone is not sufficient. A large reduction of noise level can further be obtained by operating the propeller at a low tip speed.

Reduction of Flying Skill

After the lightplane has been developed to a point at which it can be considered really useful, steps should be taken to make flying more simple. However, if development is not in the direction of making airplanes more useful, very little is to be gained by making them easier to fly. One of the attractions of flying at present is the fact that considerable skill is required, and when this element is removed there remains one less incentive to fly.

If the airplane can be made useful, the important feature in making flying simpler is not to reduce the amount of flying instruction required, but is to make flying safer. After all, if a man is going to do a considerable amount of flying, the proportion of time required for instruction is trivial, regardless of the characteristics of the airplane. However, since most accidents are caused by the pilot and not the airplane, any simplification reduces the possibility of error and, therefore, the possibility of accident. That such airplanes can be designed without further development work is demonstrated by the practical operation of the Stearman Hamond, the Ercoupe, and the General Skyfarer airplanes.

Excerpts from the paper of the same title by Otto C. Koppen, Massachusetts Institute of Technology, presented at the Annual Meeting of the Society, Detroit, Mich., January 12, 1942.

Dimensional Value of LUBRICANTS in GEAR DESIGN

by J. O. ALMEN

Research Laboratories Division, General Motors Corp.

THERE have been considerable changes in recent years in the role of lubricants as engineering materials. In times past the designer of a machine had about the same regard for the lubricant that was to be used to lubricate its moving parts as he had for the paint that would be used to decorate its exterior. Both of these questions could be settled safely after the machine was built since neither had any bearing upon the design details. Today, however, as a result of the modern demand for more intensive use of structural materials, many design details are dependent upon specially compounded lubricants. These lubricants have thus become inseparable parts of the design and the designers of lubricants share with the engineer the responsibility for the success of the machine. In these modern high-output machines the added burdens that are placed upon the lubricants vary with the kind of machine part, with the nature of the increased load, and with the kind of service that is required. Since these requirements change from time to time, the oil technicians have had to set their pace to meet the changing demands.

While it is true that we live in a continuously changing world, it is also true that really new discoveries occur very infrequently. According to available records, the first use of the principles upon which our present powerful gear lubricants are based date from the middle of the last century, and they have been in continuous use since then in cutting oils. During this time many contributors have added odd bits of information that, in total, have proved very helpful in giving us a clearer understanding of our modern gear lubrication problem. Tomlinson showed that really clean surfaces of like material, such as glass, will weld in small areas when rubbed against one another even at light loads, but that they will not weld except at high loads if the surfaces are not freshly cleaned to remove contaminating surface films. Experiments had indicated that certain oils, such as lard oil, left strongly adherent films on the surfaces of metals. The film left by perspiring hands of workmen prevented successful scraping of iron surfaces such as lathe beds. Dry sulfur was often used to treat overheated bearings. Blacksmiths required special coal for use in the forge because they had found that they could not weld iron if the coal contained sulfur. Likewise a forge could be poisoned for welding by dropping a few bird shot in the fire. White lead was commonly used as a lubricant for lathe centers. Successful cutting oils contained lard or similar oils or high sulfur mineral oils or both. These and similar bits of information were the

[This paper was presented at a meeting of the Chicago Section of the Society, Chicago, Ill., April 7, 1942.]

AUTOMOTIVE rear-axle gears present one of the most difficult gear-design and lubrication problems that are known, Mr. Almen emphasizes in this paper. The requirements are exacting, he explains, because (1) the weight must be held to a minimum; (2) the cost must be low; (3) the dimensions must be small; (4) the heat input is great; (5) the temperature is high, and (6) hypoid gears with their greater sliding velocity must be used.

Early in his paper Mr. Almen establishes the hypothesis that lubricant failure in gears and other highly loaded surfaces results in welding of small areas of the mating surface for the following reasons

1. The rubbing action removes weakly adhering films permitting contact between clean metal surfaces.

2. The temperature of a thin surface layer is very high from the friction of sliding under high load.

3. Welding of two surfaces can occur at temperatures considerably below the melting point of a metal if the pressure is great.

4. Welding will not occur if a sufficiently tenacious contaminating film is formed on the rubbing surfaces.

Explaining his title – "The Dimensional Value of Lubricants in Gear Design" – Mr. Almen points out that, as a result of the modern demand for more intensive use of structural materials, many design details are dependent upon specially compounded lubricants, and these lubricants have thus become inseparable parts of the design.

Considerable data, including three-dimensional charts, are presented to show the effect on the scoring limit of the lubricant, the pressure, the temperature, the sliding velocity, and the hardness.

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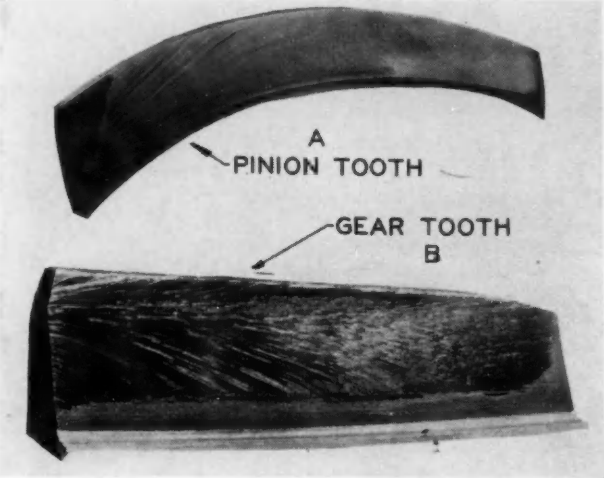
THE AUTHOR: J. O. ALMEN has long been active in General Motors research development work. He is head of the Mechanical Engineering Department No. 1 of the GM Research Laboratories.

seemingly slender clues that were available to the lubrication detectives when the need arose to explain how a lubricant must function successfully to lubricate rapidly sliding steel surfaces under pressures in excess of 300,000 psi, such as occur between gear teeth.

When hypoid gears were first introduced by the Gleason Works for Packard cars, it was found that the ordinary gear lubricants were not satisfactory and only one lubricant made to a secret formula by a single manufacturer could be used. This limited source of supply was not a serious obstacle to the use of hypoid gears so long as they were limited to one make of car but, when it appeared probable that hypoid gears would some day be used by all automobile manufacturers, it seemed desirable to learn the secrets of hypoid gear lubrication.



■ Fig. 1 – Scoring of gear teeth

As just stated, the available information on the general problem was a number of disconnected observations that had accumulated over a long period of time together with the fact that certain lubricants would operate satisfactorily under conditions where normal lubricants would fail. Failure of lubricants in gears and similar service was always characterized by surface scratches which could not be attributed to foreign matter in the oil, and occasionally, projecting points could be found on the rubbing surfaces. This result suggested that, under operating conditions of very high pressures, accompanied with rapid sliding and the generation of heat by friction, the rubbing surfaces were instantaneously welded together in small areas. The welded areas were then torn apart by continued sliding with material being pulled out of one surface and adhering to the mating surface. These projections produced the scratches and, in the process, were themselves worn away.

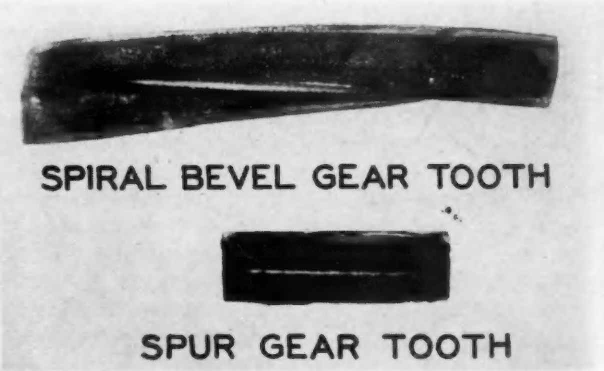
A long series of tests on the road and in the laboratory showed that lubricants containing various materials, among which were certain animal and vegetable oils, sulfur, lead and chlorine, were superior to normal mineral oils in their ability to carry high loads without scoring or scratching. This finding was in agreement with the welding theory and with the experience of the blacksmith with sulfur coal which prevented welding. The hypothesis was then formulated that lubricant failure in gears and other highly loaded surfaces results in welding of small areas of the mating surfaces because: (1) the rubbing action removes weakly adhering films permitting contact between clean metal surfaces; (2) the temperature of a thin surface layer is very high from the friction of sliding under high load; (3) welding of two surfaces can occur at temperatures considerably below the melting point of a metal if the pressure is great and; (4) welding will not occur if a sufficiently tenacious contaminating film is formed on the rubbing surfaces.

With this hypothesis as a guide, many lubricants were tried that were specially compounded to produce films on the metal surfaces that were strongly adherent and that had anti-welding properties. It was found that, in such lubricants, under high load, the strongly adhering chemical film was the actual lubricant and that the oil acted principally as a coolant. The only difference between these experiments and the experience of the blacksmith was that, in the case of hypoid lubricants, the anti-welding film was formed under the heat of friction while the blacksmith produced the anti-welding film directly from sulfur by the heat of the forge.

Of course, the ability of a lubricant to form an anti-welding film is only one of many criterions for a satisfactory hypoid gear lubricant. Other requirements, such as stability, corrosion, activation temperature, compatibility with other lubricants, and so on, have kept the oil technicians busy for many years and promise to keep them busy for many years to come.

Meanwhile the engineers also have been busy with the same problem but have attacked it from a different angle. If gear scoring is a function of the unit pressure, the sliding velocity, the temperature and the surface hardness of the mating surfaces, as has been indicated, it should be possible to vary the magnitude of these factors by design and thus possibly reduce the burden on the lubricant or alternatively permit the use of smaller, lighter, and cheaper gears. It is probable that whatever design progress is made will be utilized in the latter manner because of the economic necessity of taking more work out of each ounce of structural material. The partnership between the lubricant designer and the machine designer is, therefore, likely to draw closer with each progressive step taken by either partner.

Progress in mechanical design has been slow because of the many difficulties that have prevented full understanding of the factors that are involved. Before such a problem can be attacked intelligently, it is necessary to assemble accurate data on the effect of each of the many gear de-



■ Fig. 2 – Instantaneous contact area between teeth

sign variables. Since gears may fail to give satisfactory service for other reasons than scoring, such as tooth breakage, tooth pitting and wear, the designer must not favor the factors that influence scoring to the detriment of these other causes of failure. The only source of dependable data on gears, as on any other machine part, is actual service experience. The most useful service data come from failures and not from successes because only failures define the capacity limits of the strength of machine parts or of a lubricant. Service failures of any kind must, of course, be infrequent and, since a large number of factors are involved, the accumulation of sufficient data on any one factor, to be significant, requires a great deal of time and effort.

Many attempts have been made to devise laboratory apparatus from which basic quantitative information may be assembled more easily than from service experience, but such apparatus, no matter how elaborate, has never been successful in yielding reliable data. In general, laboratory test machines can be useful for some qualitative measurements only, since it is not possible to reproduce in the laboratory all or even a substantial part of the operating conditions that occur in service.

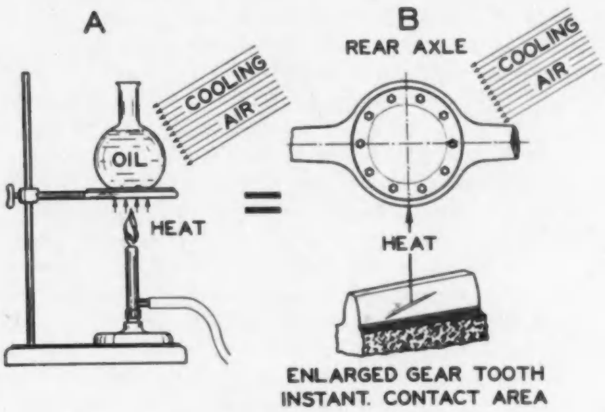
There are now in use in many laboratories a large variety of oil testing machines that not only do not agree with service experience, but fail just as badly to agree with one another. This condition is not surprising when the complexity of the problem is understood. Any testing machine that properly proportions each of the many dimensional and operating variables will have to be constructed and operated just like the machine in which the lubricant is to be used. For automobiles or trucks, this means that the testing machines will have to be complete, full-scale automobiles or trucks operated on the road. But even this is not enough because a test driver will not encounter all of the driving conditions that occur in service, nor will one or a dozen automobiles or trucks be representative of all of the automobiles and trucks that must be lubricated.

The many different oil testing machines that are used often serve to confuse rather than to clarify the lubrication problem, since the operating conditions of almost any such machine can be altered so as to show any one of several different lubricants to be superior to any other of the group. In comparing lubricants, any oil testing machine should be used as a limited grading device only. Suppose that lubricants *A, B* and *C* have been found, through service experience, to rate in that order of excellence in preventing the scoring of gears. It is then usually possible to adjust the operating conditions of any machine to rate these lubricants in the same order, but it is not safe to assume that a fourth lubricant, *D,* on which no service data are available, will be properly graded by this machine.

An increasing number of laboratories are installing dynamometer equipment for laboratory testing of full-scale axles in the hope of obtaining more reliable data than are afforded by testing machines. Such tests are perhaps more reliable than arbitrarily designed machines, but it must be remembered that an axle in the laboratory is still only another form of testing machine for which the operating conditions can be varied so as to grade a lubricant to agree or to disagree with normal road service. However, when the operating conditions are adjusted so as to produce the same scoring characteristics as previously have been determined by service data, a dynamometer axle test becomes a reasonably accurate measure of scoring, but not necessarily a reliable device for measuring other lubricant characteristics. All of which brings us back to statistical data from actual service experience as the only source of reliable information for the designer of lubricants, as well as for the designer of gears.

The result of a dynamometer lubrication test on a hypoid rear axle that is in good agreement with service experience is shown in Fig. 1 in which *A* is a pinion tooth and *B* is a mating ring gear tooth. The lubricant used in this test was a satisfactory hypoid lubricant under ordinary operating conditions, but the gears were severely scored, as is shown in the photograph, when the bulk oil temperature was increased above normal operating temperature, as might occur in ascending a long steep hill on a warm day. Note that the score lines are in the direction of sliding.

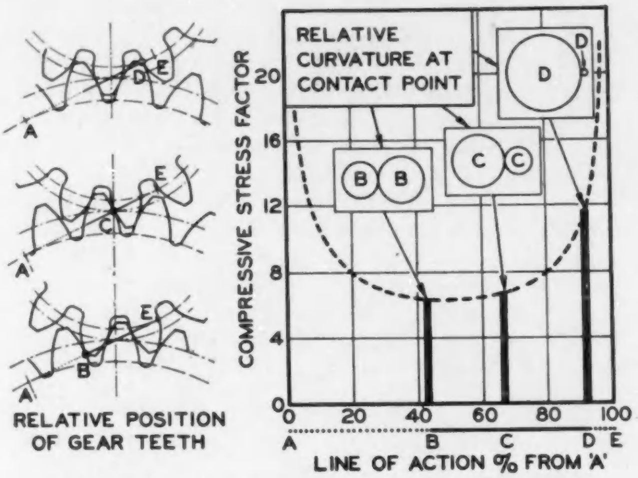
Although many serious attempts have been made, there are no means yet available for measuring the temperature of gear tooth surfaces but, that the temperature of a thin surface layer is very great, can be inferred from indirect evidence. A good mental picture of the instantaneous temperature may be had from the illustration, Fig. 2,



■ Fig. 3 – High temperatures on gear teeth due to small area of contact

which shows the size of the instantaneous contact area between mating gear teeth. All of the driving force of the gear is transmitted through this small area and, since we know the total load and the degree of curvature of each of the mating teeth, we can calculate the unit pressure. In highly loaded gears, the unit pressure may be 350,000 psi to 400,000 psi and, since these pressures occur while the teeth are sliding upon one another at high velocity, it is not so difficult to accept the welding theory of scoring. The wonder is that welding can be avoided.

A graphical representation of the rate of heat generation is illustrated in Fig. 3-*A,* in which a flask containing the oil from a rear axle is maintained at axle operating temperature by a gas flame, while air is blown over the flask surface, equivalent to the air movement over the axle while moving along the road. Obviously a considerable flame will be required to keep the oil in the flask at the axle operating temperature. In an axle the flame is



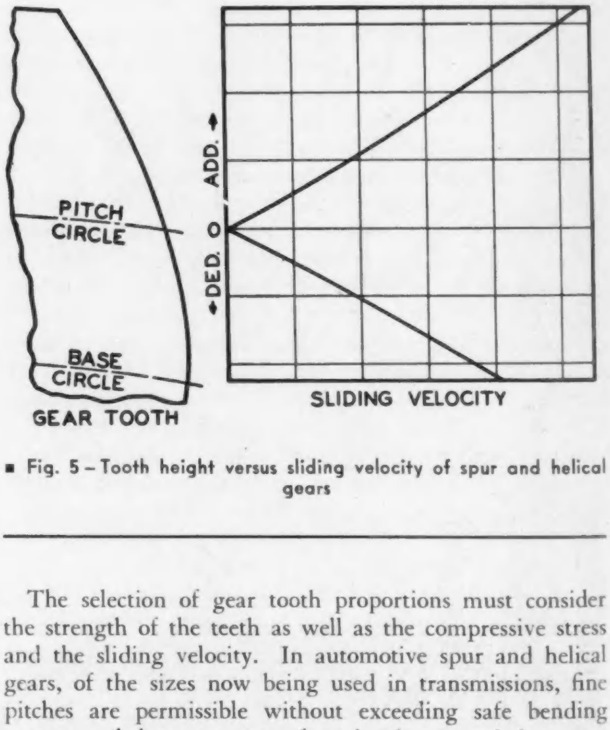
■ Fig. 4 – Influence of gear design on compressive stress

replaced by the hot instantaneous contact area as was shown in Fig. 2, which area is reproduced to a ten-times magnified scale in Fig. 3-*B*. The axle, Fig. 3-*B*, must now be maintained at axle operating temperature by the heat flow from this small area except for the additional friction heat from the ball and roller bearings. These illustrations may convey some idea of the conditions under which gears and their lubricants must perform their jobs.

The compressive stress between operating gear teeth is a function of the driving load and the relative curvature of the contacting teeth. Since the curvature varies from point to point on a gear tooth, the compressive stress also varies. Fig. 4, left, shows mating gear teeth in three different positions. In the diagram at the right are shown, by means of contacting cylinders, the relative curvatures of the mating teeth for each of the three tooth positions shown at the left. Also shown by a graph is the relative compressive stress at constant load for any point on the tooth. Note that the compressive stress, for the two-to-one gear ratio here considered, is greatest when the contact occurs near the tip of the gear tooth as at *D*. If the height of the pinion tooth above the pitch line is made somewhat longer, and the depth below the pitch line correspondingly shorter, the maximum compressive stress can be reduced without loss of tooth action. The compressive stress can also be reduced if the radii of curvature of the teeth are increased, as would result by the use of a greater pressure angle. The use of finer pitch and, therefore, shorter teeth confines the tooth action to a narrower range of compressive stress; that is, the tooth action may be designed to occupy only the lower portion of the compressive stress graph of Fig. 4. This assumes, of course, that the reduced strength of the finer teeth is permissible.

The velocity of rubbing or sliding of a gear tooth upon its mate varies with speed of rotation of the gear and the distance of the point of contact from the pitch point. The sliding velocity of spur and helical gears of the same proportion as shown in Fig. 4, is qualitatively shown in the graph Fig. 5, in which the sliding velocity is plotted against the height of the tooth. It will be seen that, from a velocity of zero at the pitch line, the sliding velocity increases as the contact moves toward the tip or the root of the tooth. It will also be seen that a decrease of the height above the pitch line and a corresponding increase of the depth below the pitch line will reduce the maximum sliding velocity without changing the tooth action. The use of finer pitch teeth is the most effective way to reduce the sliding velocity, since, the shorter the teeth, the less the sliding. We thus see that the same design change, in the particular gears shown in Fig. 4, will be effective in reducing both the compressive stress and the sliding velocity, from which it follows that the temperature also will be reduced.

For gears of constant surface hardness, the tendency to score is measured by the temperature, the compressive stress, and the sliding velocity. The temperature of any area of a gear tooth is roughly proportional to the compressive stress multiplied by the sliding velocity (*PV*) since the coefficient of friction probably does not vary greatly for similar gears in similar service. We should, therefore, expect that scoring will appear first at the tips and at the roots of the teeth because we have seen from Figs. 4 and 5 that both the compressive stress and the sliding velocity are greatest in these regions. Fig. 6 is a spur gear tooth that shows this expectation to be true. It will be seen that the scored areas are limited to narrow bands at the tip and at the root. The portion of the tooth where the sliding velocity and the compressive stress are low is not affected.



■ Fig. 5 – Tooth height versus sliding velocity of spur and helical gears

The selection of gear tooth proportions must consider the strength of the teeth as well as the compressive stress and the sliding velocity. In automotive spur and helical gears, of the sizes now being used in transmissions, fine pitches are permissible without exceeding safe bending stresses and the occurrence of scoring in transmission gears is, therefore, infrequent. In spiral bevel and hypoid rear axles, however, the loads on the gears are relatively greater than in transmissions because of their smaller relative size and greater torque. It is, therefore, necessary to use coarse pitch teeth to avoid dangerous bending stresses. With the coarser pitches come increased compressive stress and greater sliding velocities as was shown in Figs. 4 and 5.

Coarse pitch and meshing characteristics of rear-axle gears require that the addendum of the pinion teeth be made long with consequent increase in the sliding velocity

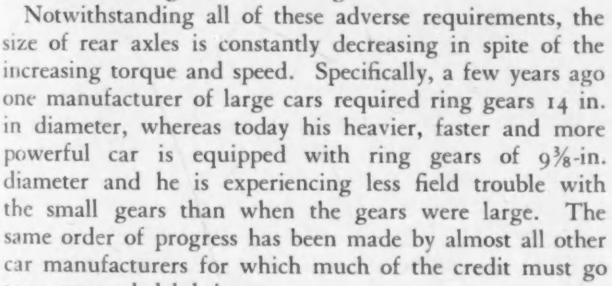
as compared with symmetrically formed teeth. Fig. 7 is a graph showing the sliding velocity in relation to the tooth height for an automotive spiral-bevel pinion and for an automotive hypoid pinion. Note that the sliding velocity is greater at the tip of the tooth than at the root for both types of pinions because of the long addenda. The velocity of sliding of the spiral-bevel pinion is zero at the pitch line but, for the hypoid pinion, the sliding does not reach zero velocity at any point and is everywhere greater than in the spiral-bevel pinion. The increased sliding velocity of the hypoid is due to the offset of the pinion by which it becomes something of a cross between a bevel gear and a worm gear. This increased sliding accounts for the greater difficulty in lubricating a hypoid axle than in lubricating a spiral-bevel axle.

Automotive rear-axle gears present one of the most difficult gear design and lubrication problems that are known. The requirements are exacting because: (1) the weight must be held to a minimum since the axle is carried as unsprung weight; (2) the cost must be low for economic reasons; (3) the dimensions must be small for road clearance; (4) the heat input is great due to high sliding velocity and high compressive stress; (5) the temperature is high due to poor heat dissipation from the housing, because of its small size, and because it is swept by hot air from the engine and hot air from the road; and (6) hypoid gears with their greater sliding velocity must be used to accommodate modern low suspension and to provide additional gear tooth strength.

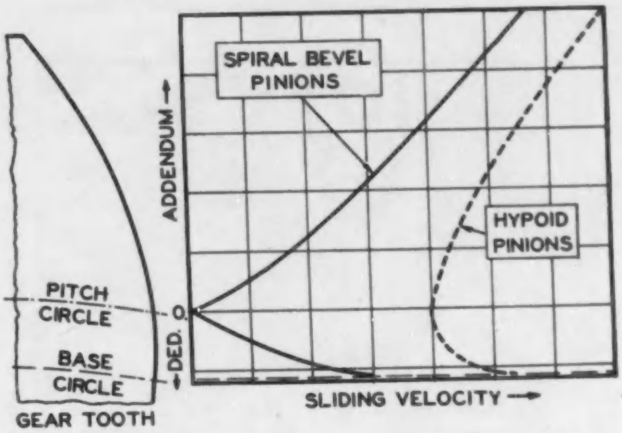
Notwithstanding all of these adverse requirements, the size of rear axles is constantly decreasing in spite of the increasing torque and speed. Specifically, a few years ago one manufacturer of large cars required ring gears 14 in. in diameter, whereas today his heavier, faster and more powerful car is equipped with ring gears of 9⅜-in. diameter and he is experiencing less field trouble with the small gears than when the gears were large. The same order of progress has been made by almost all other car manufacturers for which much of the credit must go to compounded lubricants.

The downward trend in rear-axle size is still in progress and will probably continue until an economic balance is attained between the decreasing cost of the gears and the increasing cost of the bearings. The bearing cost will probably increase due to greater loads as the gears become smaller. With the decreasing gear size will come higher tooth loads and possibly increased temperature which will have to be met by improved lubricants and by designing the gears to reduce, wherever possible, the factors that are responsible for scoring.

The job ahead will become easier as our understanding of the problem is improved. Very little data of a quantitative nature are available, but we occasionally find instances of borderline tooth scoring in which the operating gear loads and speeds are known such, for example, as was shown for a spur gear in Fig. 6. The product of compressive stress and sliding capacity (PV) to produce borderline scoring of this gear varied with the kind of lubricant that was used. For mineral oil the PV value at borderline scoring was about 3,000,000; for a mild EP lubricant the PV value equalled approximately 4,000,000; and, for the powerful hypoid-type lubricant, the PV value was approximately 5,500,000.



■ Fig. 6 – Scored gear tooth

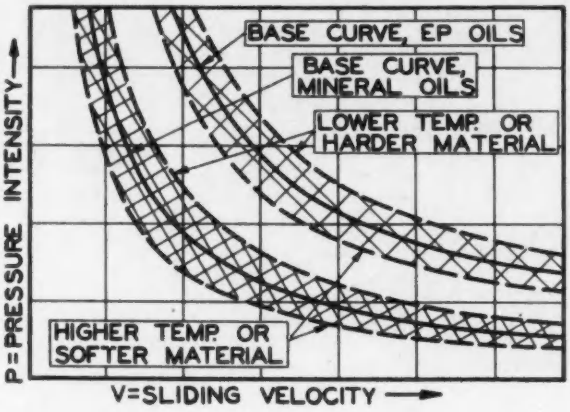


■ Fig. 7 – Tooth height versus sliding velocity of spiral-bevel and hypoid pinions

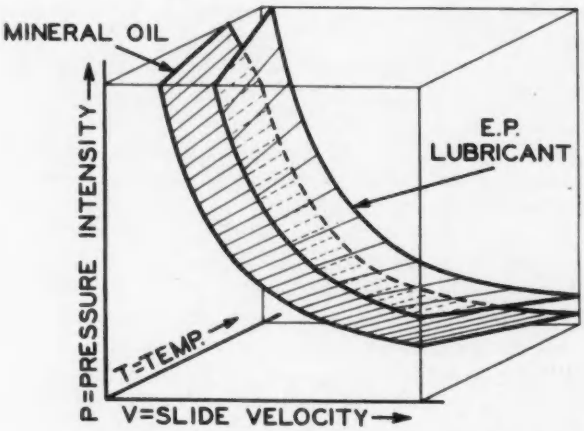
In contrast to these figures, data on spiral-bevel gears indicate permissible PV values of 1,500,000 for mineral oil and about 1,900,000 for mild EP lubricants. No PV figures are available for the more powerful types of hypoid lubricants in rear-axle gears because we do not know the relative radii of curvature of hypoid gear teeth and, therefore, we cannot calculate the unit pressures. It is probable, on the basis of other data that, in rear axles at high speeds, PV values in excess of 2,500,000 may be tolerated when hypoid lubricants are used. The discrepancy between PV values for the spur gear and spiral-bevel gears just noted will be discussed later in this paper.

The product (PV) of unit pressure (compressive stress) and sliding velocity as a measure of scoring is reasonable so long as the coefficient of friction is constant, but the constancy of the friction coefficient is questionable, especially for large changes in the magnitude of P and V. It is probable that a more general expression would take account of a decreasing coefficient of friction with an increase in velocity, and might be approximated by the expression $PV^{1/n}$. This would assume constant tooth surface temperature and constant tooth hardness.

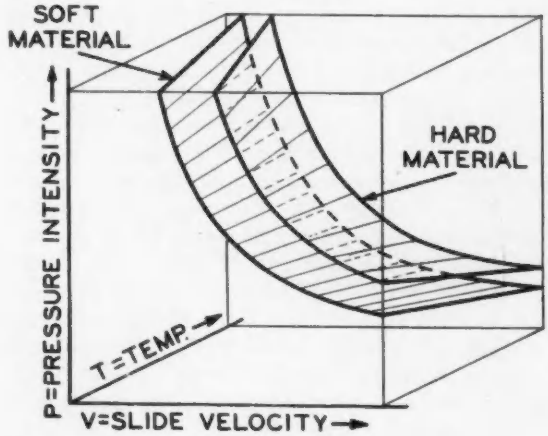
The chart, Fig. 8, shows two qualitative $PV^{1/n}$ curves which indicate the limiting values that are permissible in gears lubricated with mineral oils and with EP lubricants; that is, scoring would occur for any $PV^{1/n}$ value lying above the curve for the lubricant being considered. The position of the curves for the same lubricant would be elevated as the temperature is reduced and would be depressed as the temperature is increased as is shown by the shaded areas for each of the curves. These shaded areas will also serve to indicate that the scoring limit is elevated as the hardness of the tooth surfaces is increased and is



■ Fig. 8 – Effect of hardness, temperature, and lubricant on scoring limit



■ Fig. 9 – Effect of pressure, temperature, sliding velocity and lubricant on scoring limit



■ Fig. 10 – Effect of pressure, temperature, sliding velocity and hardness on scoring limit

depressed as the tooth surface hardness is decreased as is indicated on the chart.

Perhaps the relationships between pressure, velocity, and temperature can best be illustrated in a three-dimensional chart such as Fig. 9 in which a neutral mineral oil and an EP lubricant are plotted against these three variables. The scoring limit for each lubricant is a warped surface and the distance between the two surfaces indicates in a sense the "structural strength" of the compounded lubricant in relation to the neutral lubricant. As in Fig. 8 the position of the scoring limit surfaces will be depressed as the hardness of the tooth surface is reduced and elevated as the tooth hardness is increased.

Fig. 10 is a *PVT* chart for one lubricant only in which the scoring limit surfaces indicate the effect of hard and of soft gear tooth surfaces. This hardness effect may be an important consideration in compounding lubricants for industrial gears that are cut after heat-treatment, and it also indicates the importance of heat-treating to the maximum practical hardness.

A rather frequent occurrence of late years is the appearance of rippled hypoid gear tooth surfaces as is shown in Fig. 11. This rippling, which strikingly resembles the rippled surface of wind-blown or water-washed sand occurs during gear operation at high loads and appears to be due to actual slipping or smearing of a thin surface layer. It is perhaps the result of a relatively soft and very thin surface layer overlaying a hard base material. The shearing stress of friction causes this thin, soft layer to be displaced in the direction of slide thus forming rippled ridges at right angles to the direction of slide. The soft layer may, conceivably, be the result of operating temperatures so high as to soften the tooth surface to a depth sufficient to account for the ripples. There are some indications that rippling occurs more frequently with some hypoid lubricants than with others, which condition may be due to differences in the friction characteristics.

The surface temperature, and therefore the scoring limit of the contacting areas of gear teeth, is influenced by the degree to which the surfaces can be cooled in the interval between successive periods of contact. The heat that can be dissipated from the teeth depends upon the temperature of the lubricant, the viscosity of the lubricant, and the manner in which the lubricant is flushed over the gear teeth. The viscosity should be as low as is practical from considerations of leakage. The lower the viscosity, the more efficient it becomes in transferring heat from the tooth surfaces to the bulk oil and also from the bulk oil to the housing and thence to the atmosphere. The quantity of oil in the housing should not be greater than is required to flush the gear teeth since excess oil increases the temperature by churning and by trapping between the gear teeth. Wherever possible the lubricant should be kept away from the teeth as they go into mesh. The only effective lubricant at the point of contact is the thin film adhering to the surfaces of the teeth. More lubricant than is required to wet the surfaces merely increases the temperature.

Ideal gear lubrication would consist of a dry housing and the provision of jets to spray the lubricant over the teeth just after they come out of mesh. In this manner the surface heat can be flushed off the teeth before it has time to soak deeply into the metal and it assures that the excess lubricant is thrown off the teeth by centrifugal force

by the time they again come into mesh. The cost of such a system would, of course, be prohibitive in most automotive installations, but can be afforded in large industrial and marine gear sets.

Gear lubricants should be regarded: (1) as coolants; (2) as chemical agents for building anti-welding films; (3) as structural materials in the sense that they are important factors in determining the size of the gears and; (4) as lubricants in the usually accepted sense of a liquid film separating the two rubbing surfaces. The many volumes that have been written on the subject of lubrication make only slight reference to the lubricating conditions that prevail in gears, probably for the very good reasons that have so frequently been indicated in this paper – namely, that there are so very few data upon which to build.

When gear teeth that have seen considerable service are examined, we find little reason to suspect that the teeth have not been uniformly loaded over their entire working surfaces. Of course, we really mean that such narrow instantaneous contact bands as were shown in Figs. 2 and 3 have uniformly utilized the entire width of each tooth and have moved over the height of each tooth during each engagement. This action, however, does not actually occur because all of our structural materials are elastic and they, therefore, bend or deflect when any load is applied.

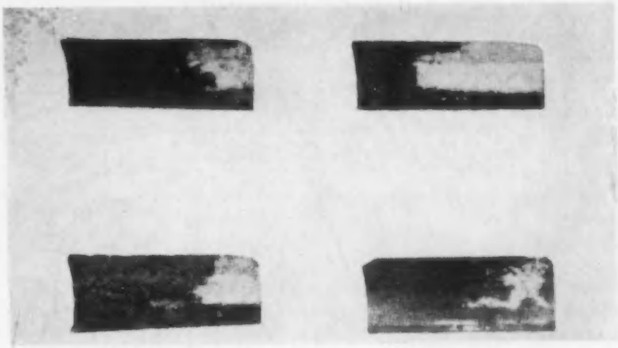
If any pair of mating teeth is exactly parallel at one load it cannot be parallel at any other load because, as the load is changed, the amount of deflection of the housing, the bearings, the gears and the shafts will change. These mating errors cannot be compensated for in cutting the gears and we must, therefore, accept as inevitable the partial use of the tooth width provided by the designer. The reason that a used gear appears to have uniformly utilized the entire width of the teeth is because the gear has experienced all manner of loads from light to heavy and it, therefore, will show the composite results of all load conditions.

When the swept area of contact for one load only is shown, as in Fig. 12, in which Scotch-tape transfers taken from transmission gear teeth reveal the actual swept area for that load, we see that the tooth width is only partially used. At very high loads the contact may extend over the whole width, but the pressure will rarely be uniform.

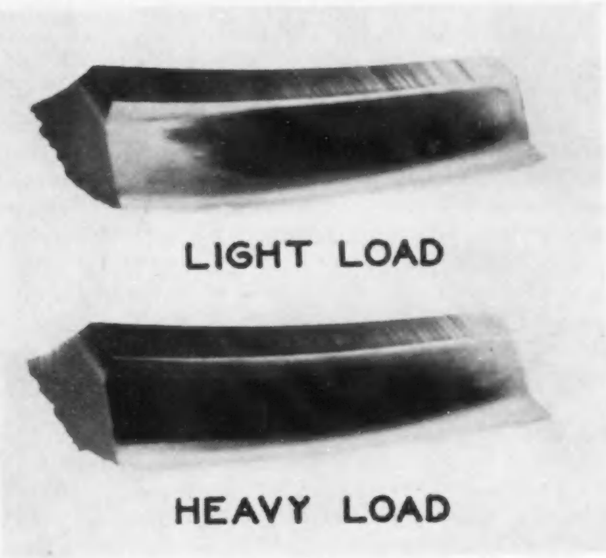
The same deflections occur also in spiral-bevel and hypoid gears but to a greater degree. In such gears, however, the teeth are cut so as to mismatch and thus provide a considerable compensation for the mating errors that would otherwise be intolerable. By mismatch is meant that the concave surfaces of the gear teeth are cut to slightly greater radii than the convex surfaces of the mating pinion teeth, and the mating teeth may, therefore, rock upon one another as loads and deflections change, like the rockers of a rocking chair upon the floor. This action is shown in the swept-area photographs, Fig. 13, which show the contact area for light loads and for heavy loads. Note that a portion of the large end of the tooth at light load is not making contact and that, at the high load, a portion of the small end of the tooth is not making contact.

While this method of cutting does not permit the utilization of the whole tooth width, it is greatly superior to teeth designed to make contact over the entire width because, in the latter case, under changing loads and there-

▪ Fig. 11 – Rippled hypoid pinion



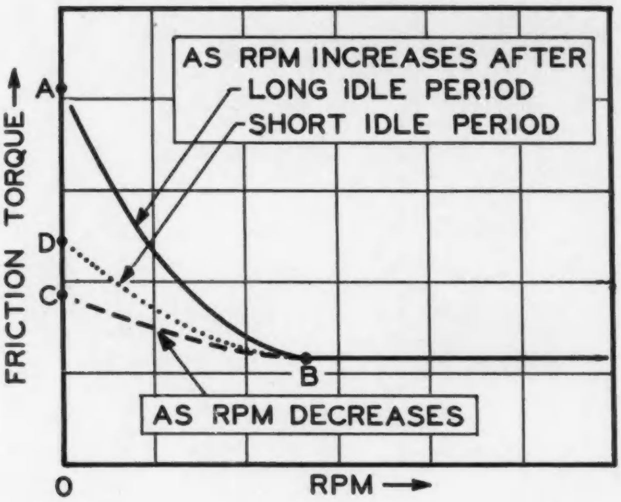
▪ Fig. 12 – Gear contact impressions



▪ Fig. 13 – Contact impression of hypoid gear

fore deflections, the contact could occur only at the extreme ends of the teeth. The amount of mismatch that is needed depends upon the rigidity of the design or upon the compensating deflections of the design. Since the lubrication difficulty varies with the unit pressure and, therefore, with the degree of mismatch, the elastic characteristics of the housing, shafts, bearings, and gears become part of the lubrication problem.

That elasticity can be useful as well as harmful is shown in Fig. 6 in which uniform load distribution over the



■ Fig. 14 – Friction torque versus rpm between bearing balls and races

width of the tooth is indicated by the uniform width of the scored bands. In this gear, the toothed rim was attached to the hub through a thin central web or diaphragm, whereby lack of parallelism between mating teeth was adjusted. A load applied at one end of the tooth would cause the diaphragm to deflect and closely approach parallel relationship between the mating teeth.

It will be recalled that the permissible *PV* values for the gear tooth shown in Fig. 6 were about twice as great as the permissible *PV* values of spiral bevels. This discrepancy is at least partially explained by the more uniform load distribution over the tooth width as has just been discussed.

The persistence of an oil film under high load is illustrated strikingly by an experiment in which a ball bearing was used as a friction-measuring device. The bearing contained only three balls and was constructed so accurately that it could be run for considerable periods without the usual ball separator. Since the separator was omitted, the readings were the friction between the balls and races only in contrast to the usual ball-bearing friction tests which include the friction between the balls and the separator. The bearing was loaded by means of a calibrated spring to a maximum pressure of 240,000 psi between the balls and the races.

The friction torque was measured at small speed increments including the starting torque with the results shown in Fig. 14. From the starting torque at *A* the friction diminished rapidly with increasing speed until it reached a constant value as at *B*. With decreasing speed the friction torque followed the line *BC* and, if immediately started again, the friction retraced the line from *C* to *B*. If after stopping, the apparatus was not again started for several seconds, the friction would follow the line *DB* as the speed was increased and then retrace the line *BC* with decreasing speed. The original starting torque *A* would not be re-established until the apparatus was permitted to stand for a considerable period before being started again, whereupon the friction would follow the original line *AB*. This result seemed to indicate that a relatively long time was required to squeeze the oil from between the balls and the races with pressures up to 240,000 psi.

This persistence of the oil film at high pressures seemingly contradicts statements previously made in this paper that the lubricant plays but a small part in separating gear tooth surfaces. However, the thickness of an oil film under the foregoing conditions is extremely small in comparison with the surface irregularities of the most highly finished mating surfaces. Metallic contact probably occurred under all of the test conditions enumerated, but the oil trapped in the surface irregularities served to reduce the intensity of the pressure on the projecting points and thus reduced the friction. Note the surface roughness of the gear tooth shown in Fig. 6 and the absence of wear on the unscored portion of the tooth after approximately 25,000,000 meshing cycles at unit pressures on the order of 200,000 psi.

The viscosity of the lubricant used on this gear conformed to SAE 10 specification. However, the actual viscosity of the lubricant trapped in the contact area was probably quite different. The high temperatures of the contact area would tend to reduce the viscosity, whereas the high pressure would cause the viscosity to increase. We do not know the temperature or the pressure-viscosity characteristics of the lubricant and we can, therefore, only speculate on the actual viscosity. Since the *PV* values and, therefore, the instantaneous temperature increase range from zero at the pitch line to a maximum at the tip and root of the tooth, it may well be that the viscosity was high in the central portion of the tooth and relatively low at the tip and the root. This condition would be in substantial agreement with the markings on the tooth.

■ Product of Many Industries

Automobiles, like most other things that we regard as necessities, would not be available to so large a percentage of our population but for the cooperative efforts of many industries and their technicians. Through industrial team work the cost of the finished product comes within the financial means of the many. The modern automobile is the joint product of the steel manufacturers, the machine-tool builders, the paint makers, and so on through a long list of industries. In no case, however, is the partnership so close as between the petroleum industry and the automobile manufacturer because both come in direct contact with the consuming public and both are directly responsible to the consumer for the products which they sell. The size, cost, and performance of the automobile engine is directly related to the knock characteristics of the fuel supplied by the petroleum industry. Similarly, the size, cost, and performance of hypoid gears is measured by the available lubricants.

SIX-WHEEL BRAKING and Its Related Problems

by W. F. BENNING and M. C. HORINE

Mack Mfg. Corp.

SIX-WHEELERS offer no greater problem in the attainment of maximum safe and controllable braking than do other types of heavy vehicles. That difficulties have been experienced and that complaints have been registered admittedly indicates that, in many cases, the problem has not been solved correctly. Analysis of the dynamics of braking, however, indicates that, far from presenting a more difficult problem than do other types, the six-wheeler can be made to give better braking performance than any.

Specifically, the complaints concerning the behavior of six-wheel brakes involve what is known as bogie hopping and loss of steering control when brakes are applied on slippery roads. On the credit side of the ledger, six-wheelers are generally credited with being less liable to skidding than either four-wheeled straight trucks or tractor-semi-trailers. In respect to general stopping ability, also, six-wheelers rate high.

The first of the defects complained of, namely bogie hopping, is peculiar to rigid six-wheeled vehicles and multi-axle trailers and so merits particular study by builders of these types. The steering difficulty, however, is found in both rigid and articulated types of multi-axle vehicles, that is, in both six-wheel straight trucks and in tractor-semi-trailers. Nevertheless, it is said to assume its severest form in the rigid six-wheeled truck.

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FAR from presenting a more difficult problem than other types, the six-wheeler can be made to give better braking performance than any, these authors contend. They are less likely to skid than either four-wheeled straight trucks or tractor-semi-trailers, they continue, and in general stopping ability they rate high. However, they acknowledge the complaints against the six-wheeler for behavior known as "bogie hopping" and loss of steering control when brakes are applied on slippery roads.

Comparing the three principal types of commercial vehicles, they conclude that the complaints on this score are based if at all upon something other than dynamic weight transfer.

"The reason why the problem is so acute with six-wheelers is that the supposedly rigid alignment of the rear bogie wheels constitutes a resistance to steering and that, therefore, any tendency for the front wheels to lose traction is of graver concern than with either a four-wheel truck or a tractor-semi-trailer." Calling this the crux of the six-wheel steering problem so far as brakes are concerned, they declare: "for six-wheelers, therefore, a relay valve to speed up the application of the rear brakes and a quick-release valve for the fronts seem essential," while a brake distribution quite different from that of the four-wheeler is required.

"Bogie hopping," they say, "as in the case of steering ability, is affected by weight transfer," and, according to the authors, it is not an inherent fault of six-wheelers, but "merely a manifestation of faulty design which can be completely overcome by employing a bogie which is torsionally balanced."

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Braking Fundamentals

At the outset it is well to recognize that, short of failure to stop within a reasonable distance, practically all irregularities of braking arise from improper distribution of braking force among the wheels and, in all such cases, the evils arise from the untimely locking of one or more wheels. Such irregularities cannot be disposed of by merely cutting out or reducing to impotence the offending brakes, for this will reduce the total retardation in a corresponding degree. It seems elementary to point this out, but the point is so often lost sight of that a reaffirmation may be pardoned. In fact a review of several fundamentals of braking appears to be a necessary prelude to an analysis of the special problems of six-wheel braking.

Retardation by means of friction brakes acting upon the wheels of road vehicles is limited by two principal factors, namely the coefficient of adhesion between the tires and the road and the degree to which the distribution of braking force among the wheels coincides with the imposed load on each wheel. The first factor is usually disposed of by assuming for pneumatic tires on average improved roads a coefficient of 0.6. The second is much more complicated and difficult to ascertain.

Adhesion Coefficient

Of course, the assumed constant of 0.6 is in reality not a constant at all. The adhesion of the tires constantly changes. Not only does it differ as between different types of paving, but at different points along any given roadway, while rain, snow, sleet, mud and ice produce tremendous changes. Tires differ and, as the treads wear away, they change. However, over the distance required to make a normal stop, the changes in adhesion may be assumed to affect all of the tires on a vehicle approximately alike and so will not materially upset the dynamics of braking.

The limitation imposed by the 0.6 coefficient is by no means an embarrassment, for it permits a rate of retardation of 19.32 ft per sec^2 which, if sustained uniformly, is equivalent to a braking distance of 22.2 ft to stop from 20 mph. However, since perfect brake distribution is by any means now known unattainable, this rate of deceleration likewise is unattainable on prevailing roads.

In perfect braking each brake would exert a retarding force just below that at which the shoe would seize the drum and cause the wheel to slide. Any lesser force would reduce the rate of retardation. Any greater force would cause all wheels to slide simultaneously, thus reducing the rate of retardation again and endangering the directional stability of the vehicle as well.

Braking Distribution

Any deviation from this perfect balance of braking force against imposed load on each wheel will cause one or more wheels to lose traction and slide before the others do. If the sliding wheels are the steering wheels, then control of steering will be jeopardized. If the sliding wheels are on one side, the vehicle will have a strong tendency to skid. In fact every vagary of imperfect braking can be traced back to imperfect coincidence of braking force and imposed load.

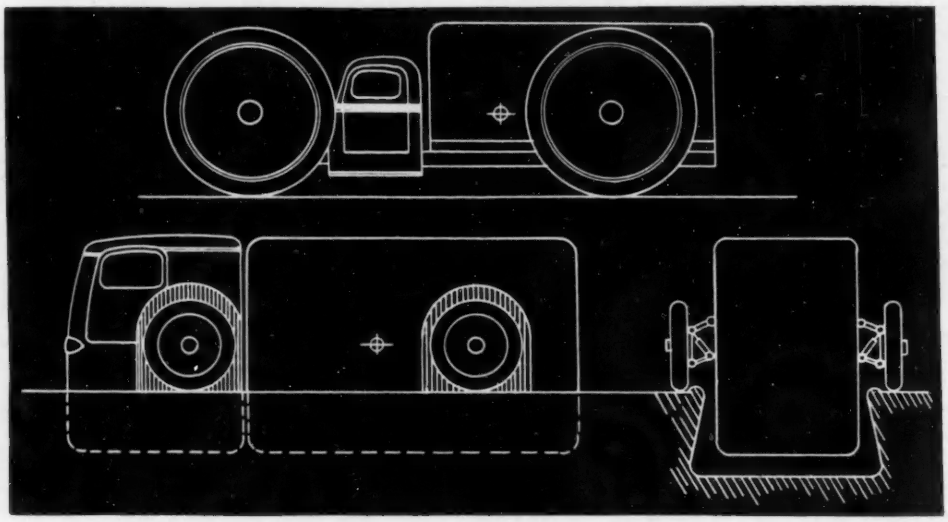


Fig. 1 – Two methods of locating the center of gravity of a vehicle at the same level as the wheel spindles

If only static weight distribution could be maintained with the vehicle in motion, the attainment of perfect braking balance would be conceivable, or even, if the transfer of weight from axle to axle were always the same under maximum brake application, a far closer approach than is now in sight could be reached.

Static and Dynamic Weight Shifting

Not considering transverse shifting of weight, due to road crown, unequal distribution of the freight in the body, sidesway and centrifugal force in rounding curves, the transfer of weight longitudinally results in a constant shifting of weight from axle to axle. The influences affecting this shifting are both static and dynamic. Under the static classification are differences in the weights of payloads, unequal longitudinal distribution of the freight, differences in the density of freight, and road gradient. Under the dynamic classification are the inertia of the vehicle and its load under acceleration and deceleration and the fluidity of the load itself, the jouncing of the springs and tires, and the torque reactions of driving and braking.

Maximum braking effectiveness requires that each wheel shall exert a retarding force exactly equal to 0.6 times the actual weight imposed upon it. If this imposed weight were constant on each wheel, this would be a simple matter of adjusting brake-chamber sizes, line pressures, cam contours, or lever lengths to produce perfect balance; but, even if the static weight distribution were constant, the dynamic forces would unset such uniformity. The dynamic effects, however, could be neutralized if the center of gravity could be located in the same plane as the wheel spindles and if axle torque were balanced out.

More careful loading of vehicles would help. Elimination of all grades would simplify the problem. Equal distribution of both tare and payload weight front and rear would eliminate one more variable. Fig. 1 illustrates two methods by which a vehicle might be designed so that, under full load of given density, the center of gravity could be kept at the same level as the wheel spindles.

But every one of these things is as impracticable and unlikely of achievement as the constructions shown in Fig. 1 are impractical.

Effects of Grades

However, even such fantastic vehicles as these would not be immune to the effect of grades for, regardless of all else, a down grade will surely cause transfer of weight to the front axle and an upgrade will transfer it away from the front axle. So far as brakes are concerned, this is fortunate, because it is in descending grades that brakes assume their greatest importance and control of steering becomes most vital. No matter how the braking distribution is laid out, the danger of front-wheel locking must of necessity decrease on downgrades. Loss of steering control is most to be feared on upgrades when brakes are applied.

A fact not generally realized is that, while grades cause a gain in weight on the axles on the low side and a loss of weight on those on the high side, this gain and loss do not correspond, so far as pressure of the tires on the road surface is concerned. In other words, the total ground pressure decreases with the grade. Thus, on an upgrade:

$$\text{Rear ground pressure} = \left(\frac{F}{WB} + \frac{H}{WB} \times \tan \alpha\right) \times GVW \times \cos \alpha$$

$$\text{Front ground pressure} = \left(\frac{R}{WB} - \frac{H}{WB} \times \tan \alpha\right) \times GVW \times \cos \alpha$$

Where F is the horizontal distance from the center of gravity to the point of contact of the front tires with the road;
R is the horizontal distance from the center of gravity to the point of contact of the rear tires with the road;
H is the height of the center of gravity above the road;
GVW is the gross vehicle weight
α is the grade in degrees from the horizontal

This condition, of course, is because a part of the weight becomes a force tangential to the tires and parallel with the road surface. To see why this is so it is only necessary to consider that, if the vehicle were at a 90-deg angle grade, there would be no pressure between the tires and the road.

This is not cited as a mathematical curiosity, but to point out the important fact that brake effectiveness decreases in some proportion to the grade, reaching zero when the loss of weight normal to the road surface reaches 60% of the gross vehicle weight, assuming 0.6 adhesion. This, of course, would be a 60% grade. This means that, on a grade of only 15%, a vehicle loses 25% of the braking ability it possesses on a level road, due to reduction of total ground pressure alone. In addition to this, of course, the alteration of distribution between the axles will occasion additional loss of brake ability, aggravated by further transfer due to inertia on down grades and alleviated by the same effect in going uphill. Torque reaction, too, will affect the result.

These effects are common to all types of vehicles, being most drastic, of course, where the transfer of weight is greatest, namely on the four-wheeler and least in the case of the tractor-semi-trailer. Of all of the influences affecting weight transfer on grades commonly encountered in commercial service, that of mass inertia, particularly in traversing scalloped road surfaces is the most severe.

Optimum Braking Distribution

An inconstant position of the center of gravity is unavoidable in highway vehicles and by that token any fixed distribution of braking effort can coincide with the actual distribution of weight only under certain fixed static and dynamic conditions and therefore can occur in the operation of a truck only momentarily. It is therefore of the greatest importance that the brake engineer select the particular assumed load distribution upon which his brake balance is based with the greatest of care. Obviously he need concern himself solely with conditions existing during brake applications of some severity. Whether he should consider the unloaded condition, normal load, overload, or average or mean load is not easy to decide. Certain it is that, the greater the severity of braking and the greater the load on any rationally designed vehicle, the greater will be the percentage of load borne by the front axle.

If, therefore, the brakes are balanced for the lightly loaded condition, a mild application will suffice for the maximum retardation and the shortest attainable stop will be made. Now, if brakes so balanced are used under a full load or overload, the transfer of weight to the front axle will be materially increased, so that under maximum application the front wheels will not lock before the rear ones do, if at all. This arrangement will avoid the hazard of loss of steering control, but maximum retardation will not be reached before the rear wheels lock.

If, on the contrary, the brakes are balanced for the full-load condition, the shortest attainable stop will be made with full load. Under light load, however, the transfer of weight to the front axle will be at a minimum and, consequently, too much braking force will be distributed to the front axle, thus engendering danger of loss of steering control. If the application is restricted to that at which the front wheels will not lock, then the major part of the retardation – that of the rear wheels – will be lost and an excessive length of stop will result.

Obviously, as in so many other engineering problems, a compromise must be arrived at which, for the sake of adequate braking under full load without too much danger of front-wheel sliding when light, will strive for some optimum between the foregoing extremes.

Of course, the preceding applies to all types of vehicles and so, in respect to the transfer of weight to the front

axle, they differ mainly in the magnitude of this transfer. Naturally the amount of weight transfer varies as between individual vehicles and also on the same vehicle on each trip as well as on different grades and depending upon how severely the brakes are applied.

■ Three Types Compared

Nevertheless, since six-wheelers particularly are singled out for criticism, it is necessary to make some general comparison of the characteristics of the three principal types of commercial vehicles in this respect. For this purpose we may accept as normal or representative the three following:

Weight Transfer Between Axles Due to Braking

Type	Four-Wheeler	Six-Wheeler	Tractor-Semi-Trailer
Body length, ft	15	20	20
CA dimension, in.	120	138	..
Fifth wheel, offset	..	..	6
Wheelbase, in.	194	188–48	140–182½
Static tare weight, lb	13,500	18,400	19,000
Front	6,300	6,200	5,650
Middle	..	6,100	7,350
Rear	7,200	6,100	6,000
Static payload, lb	11,500	21,600	21,000
Front	1,600	1,600	400
Rear	..	10,000	9,100
Middle	9,900	10,000	9,500
Static gross weight, lb	25,000	40,000	40,000
Front	7,900	7,800	6,050
Rear	..	16,100	16,450
Middle	17,100	16,100	17,500
Height center of gravity (above spindles), in.			
Truck or trailer	39	39	49
Tractor	..	..	11
Decelerating rate, ft/sec²	14.35	14.35	14.35
Inertia transfer			
Front	+2250	+3250	+ 490
Middle	..	−1625	+3010
Rear	−2250	−1625	−3500
Brake torque transfer			
Front	+1150	+ 330	+1460
Middle	..	− 165	− 630
Rear	−1150	− 165	− 830
Total transfer			
Front	+3400	+3580	+1950
Middle	..	−1790	+2380
Rear	−3400	−1790	−4330
Dynamic distribution	25,000	40,000	40,000
Front	11,300	11,380	8,000
Middle	..	14,310	18,830
Rear	13,700	14,310	13,170
Per cent on front			
Static	31.6	19.5	15.1
Dynamic	45.2	28.4	20.0

In the foregoing analysis, it will be observed that the two sources of weight transfer have been considered separately and then combined, as reflected in the following formulas, just used:

For weight transfer to front axle:

$$\text{Due to Inertia} = \frac{CGH \times GVW \times BC}{\text{Wheelbase}}$$

$$\text{Due to Brake Torque} = \frac{RR \times GVW \times BC}{\text{Wheelbase}}$$

Where CGH = Height of center of gravity above wheel spindles;
GVW = Gross vehicle weight;
BC = Braking coefficient in terms of coefficient of tire; adhesion equivalent to retardation rate of 14.35 ft/sec²;
RR = Rolling radius of tire.

In the case of six-wheelers whose rear brake torque is transmitted directly to the frame, as with four-wheelers, the above equations hold. However, where the rear bogie design is such that the brake torque is balanced out, so that none is transmitted to the frame, the formula for transfer of weight to the front axle, due to brake torque becomes:

$$\frac{RR \times FGW \times BC}{\text{Wheelbase}}$$

Where FGW = Gross vehicle weight on front axle.

The same formulas are used to determine the weight transfer of a tractor-semi-trailer by solving separately for the semi-trailer and the tractor, as though the fifth wheel were the front axle of the semi-trailer. In so doing, it must be understood that the effective center of gravity on the tractor will be the combined centers of the tractor alone and the fifth-wheel trunnion (which is the center of gravity of the dynamic weight imposed on the tractor by the trailer). Accordingly, the formulas for the tractor-semi-trailer are:

For semi-trailer weight transfer to fifth wheel:

$$\text{Due to Inertia} = \frac{CGH \times GWT \times BC}{AK}$$

$$\text{Due to Brake Torque} = \frac{RR \times RGW \times BC}{AK}$$

Where GWT = Gross weight of trailer;
AK = Axle to kingpin dimension;
RGW = Gross weight on trailer axle.

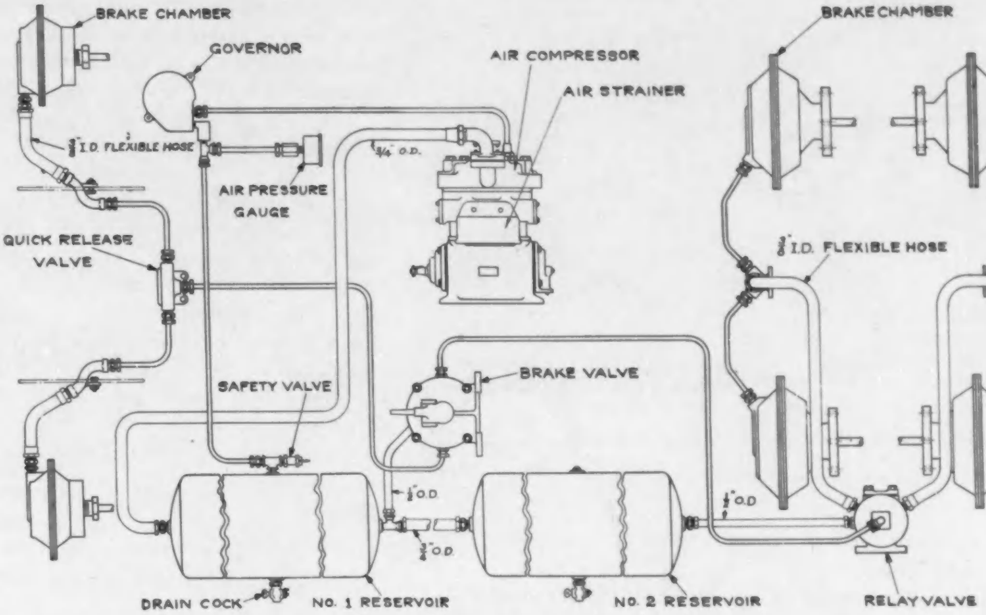
For tractor weight transfer to front axle:

$$\text{Due to Inertia} = \frac{HCG \times TGW \times BC}{\text{Wheelbase}}$$

$$\text{Due to Brake Torque} = \frac{RR \times TGW \times BC}{\text{Wheelbase}}$$

Where HCG = Height of combined centers of gravity of tractor alone and fifth-wheel trunnion;
TGW = Total gross weight of tractor plus the sum of the static gross weight of trailer imposed on fifth wheel and the weight transferred thereto by inertia and trailer brake torque.

The authors are aware that, in the preceding calculations of weight transfer due to mass inertia, a departure has been made. It is customary to consider the points of tire contact with the ground as the pivot points about which the mass centered at the center of gravity gyrates. However, it is believed to be more correct to separate the simple inertia transfer from that caused by torque reaction. So considered, it is the spindles which become the pivot points, as they certainly are with respect to the torque moments from the brakes.

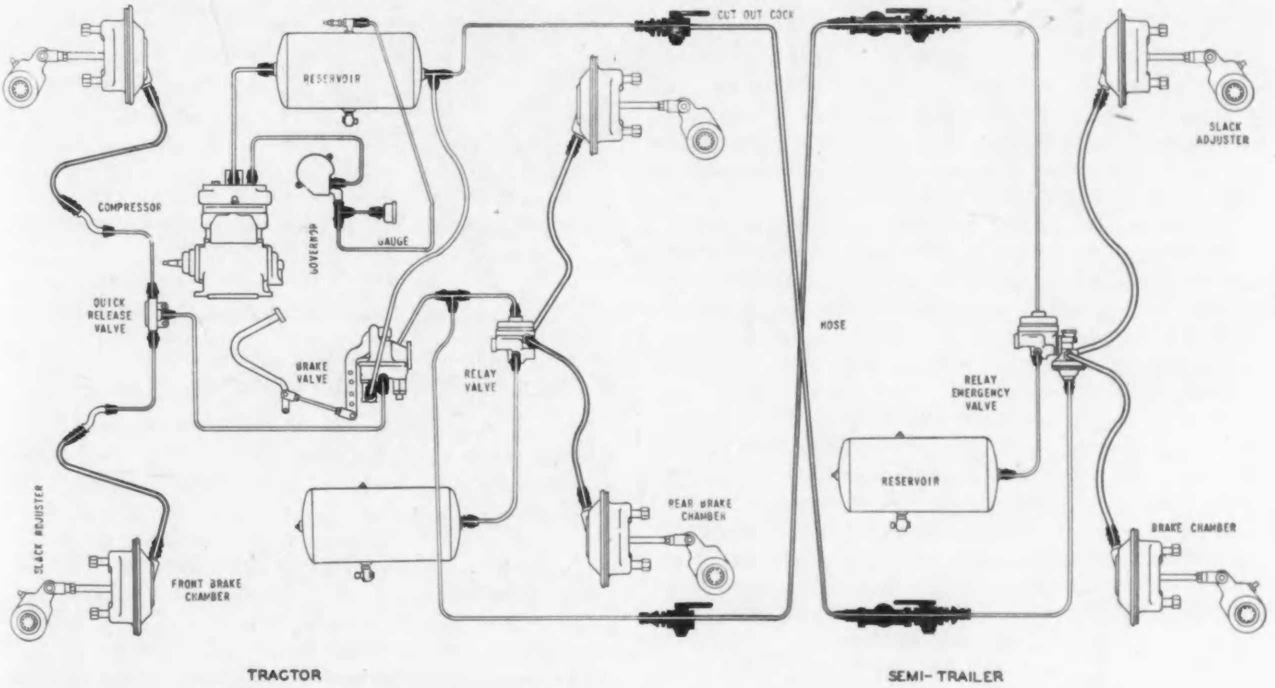


■ Fig. 2 – Air-brake hook-up for six-wheeler

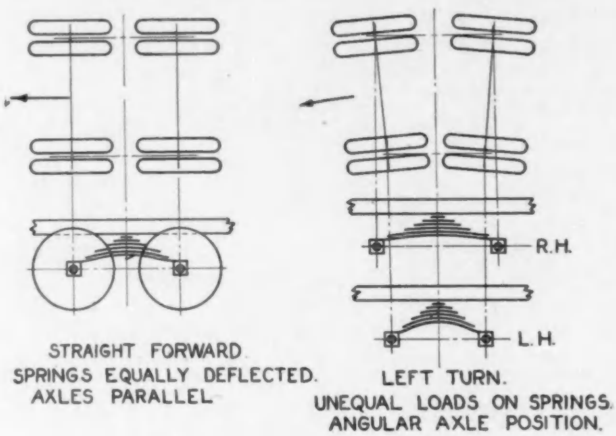
In explanation of the marked difference in center of gravity height as between the straight trucks and the semi-trailer, it should be noted that some of this difference is accounted for by the relative lightness of the semi-trailer chassis as compared with those of the straight trucks and also that the centers as given in the table are those above wheel spindle height and not as usually given, above the road level.

The rate of deceleration (14.35 ft per sec^2) represents a normal stop of 30 ft from a speed of 20 mph.

In the case of the tractor-semi-trailer, these formulas have been used separately for the trailer and then, using



■ Fig. 3 – Air-brake hook-up for tractor semi-trailer



■ Fig. 4 – Steering effect of rear axles

the transferred load at the fifth-wheel trunnion, they are used for the tractor.

A different method has been used by Heldt[1] in analyzing weight transfer of a tractor-semi-trailer, in that the two components of weight transfer, namely, mass inertia and brake torque are not separated; the pivot points are assumed to be the points of tire contact with the ground; and that kingpin offset has been ignored.

In view of the foregoing, showing that, of the three types, the six-wheeler evinces neither the greatest nor the least variation in front-wheel loading and that therefore it should offer little difference in its potentialities for front-wheel retardation without risk of loss of steering control, it must be concluded that the complaints on this score are based if at all upon something other than dynamic weight transfer.

■ Steering Reluctance of Bogie

The reason why the problem is so acute with six-wheelers is that the supposedly rigid alignment of the rear bogie wheels constitutes a resistance to steering and that, therefore, any tendency for the front wheels to lose traction is of graver concern than with either a four-wheel truck or tractor-semi-trailer.

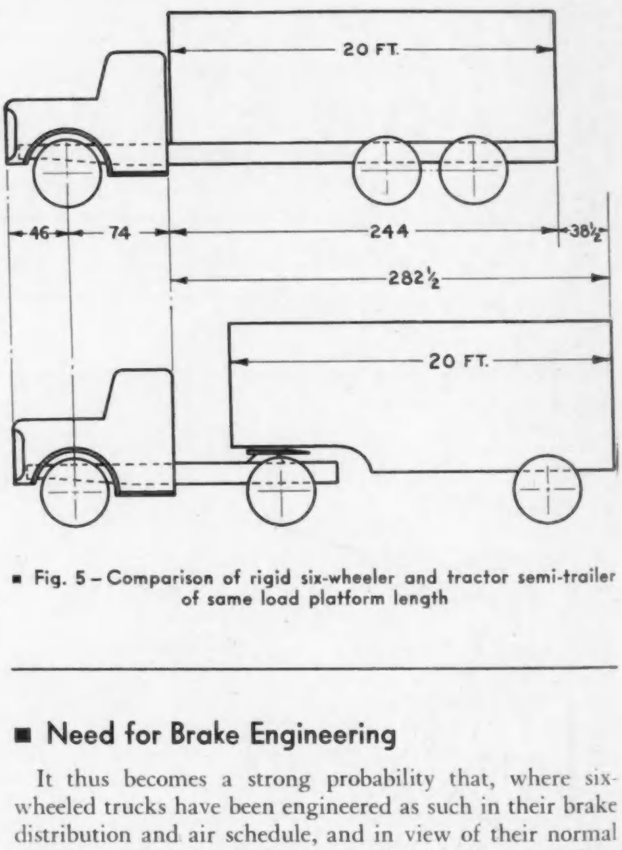
This is the crux of the six-wheel steering problem so far as brakes are concerned. In the Pacific Coast states many operators have attempted to solve the problem by eliminating front-wheel brakes on six-wheelers, advancing the excuse that, as not over 15% of the braking is on the front wheels of a six-wheeler, this portion may be sacrificed for the sake of safety.

Makers of brake equipment have come forward with another solution in the shape of the limiting valve. This valve serves to reduce the distribution of line pressure to the front wheels, so that full braking effect may be provided for the fully loaded condition and this amount reduced when under partial or no load for the sake of safety.

While we must accept as a fact the existence of a problem with respect to the steering control of many six-wheelers now operating on the highways, we must not conclude from this condition that this trouble arises from an inherent defect. It must be remembered that many six-wheelers are adaptations of four-wheeled vehicles, to which attachment third axles have been applied or which have merely had a four-wheeled rear bogie substituted for the standard rear axle with little or no change in the front brakes.

Although this paper cannot include the important matter of air-brake schedules in its scope, it is well to bear in mind that the timing of six-wheeler brakes is probably quite as important as that of tractor-semi-trailers. The simple direct-control system will not do because the relatively shorter distance from the application valve to the front-wheel brakes produces a very undesirable lead on front brake application.

For six-wheelers, therefore, a relay valve to speed up the application of the rear brakes and a quick-release valve for the fronts seem essential while, as just observed, brake distribution quite different from that required for four-wheel trucks is clearly indicated. Brake schedules for six-wheelers and tractor semi-trailers are shown in Figs. 2 and 3, respectively.



■ Fig. 5 – Comparison of rigid six-wheeler and tractor semi-trailer of same load platform length

■ Need for Brake Engineering

It thus becomes a strong probability that, where six-wheeled trucks have been engineered as such in their brake distribution and air schedule, and in view of their normal weight-transfer characteristics, much of the over-application of front-wheel brakes should disappear. With the limiting valve, adjusted in accordance with the load, the condition of the road, and according to whether the gradient is up, down or neutral, a very high degree of exactitude in brake distribution can be achieved.

The one question remaining is that of the rear bogie's reluctance to steering. That this condition is pronounced in short-wheelbased COE types and in tractors is undeni-

[1] See *Automotive Industries*, Vol. 85, No. 12, Dec. 15, 1941, pp. 40-41: "Transfer of Load Due to Braking in Tractor-Trailer Combinations," by P. M. Heldt.

ably a fact but, in the long-wheelbased types used in long-distance freighting, the longer lever arm through which the front wheels operate in relation to the bogie should easily overcome this reluctance.

■ Some Bogies Self-Steering

Further, the assumption that a bogie necessarily resists steering is not warranted. It is perfectly true that, when horizontal radius rods or reach members are used to align the axles, they will be held in parallelism but, with oblique radius rods or with positively cambered springs in lieu thereof, there is a positive steering action within the bogie itself.

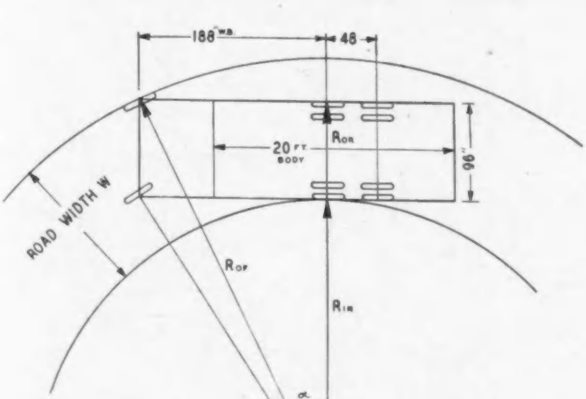
Fig. 4 illustrates the action of a bogie of the latter class. Here it will be seen that, when proceeding on a straight course, the two axles are parallel but that, when rounding a turn, centrifugal force will tend to shift the center of gravity outwardly, thus causing the outer spring to deflect and the inner one to retract. The positive camber causes a corresponding elongation of the outer spring and shortening of the inner, thus producing a positive steering effect. It has been found that this effect is so pronounced in some vehicles that the ease and positiveness of steering is of a high order.

■ Tire Wear

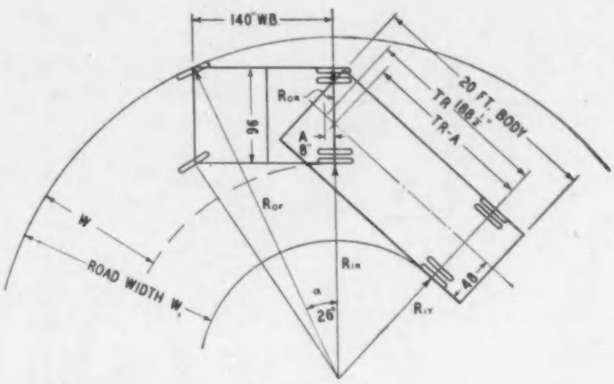
It may be noted in passing that this steering action, inherent in some bogie designs, explains why tire wear experience with six-wheelers is subject to considerable variation. In those bogie types in which the axles are constrained by their radius rods or reach members to constant parallelism, there must be considerable scuffing on turns, whereas this action is largely missing in types having a more elastic relationship between the axles.

■ Turning Radius

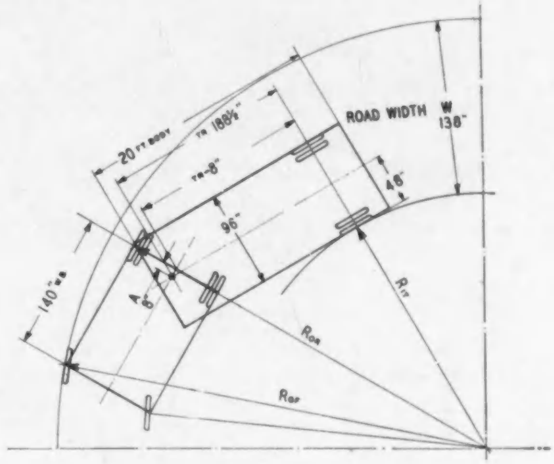
While on this subject of steering, there is another misconception respecting six-wheel characteristics that might as well be cleared up. Because of its articulation, the tractor-semi-trailer is unquestionably capable of maneuvering in closer quarters than is the rigid six-wheeler, despite its greater overall length for the same load platform length (see Fig. 5).



■ Fig. 6 – Turning radius and required lane width of six-wheeler with a 20-ft body



■ Fig. 7 – Turning radius and required lane width of tractor-semi-trailer with 20-ft body



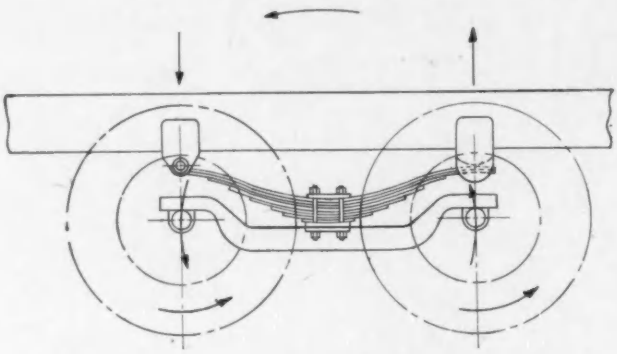
■ Fig. 8 – Minimum radius at which the tractor-semi-trailer can turn within the same 11½-ft lane width required by the six-wheeler

From this it is often assumed that the six-wheeler is at a disadvantage in negotiating tortuous highways. The facts are, to the contrary, that, while the tractor-semi-trailer by reason of the shorter wheelbase of both the tractor and trailer, is able to turn in a shorter radius, it also requires a greater width of road in which to make a turn.

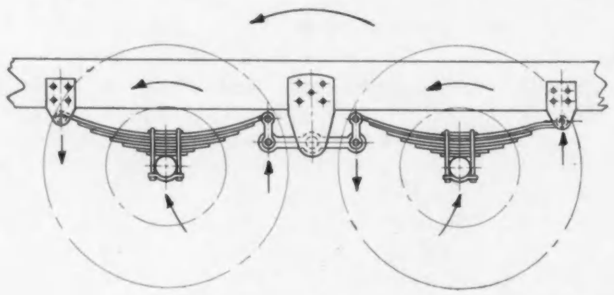
Fig. 6 represents a six-wheeler with a 20-ft body in diagrammatic form. With the front wheels turned to 33 and 26 deg, respectively, showing a turning radius of 427 in. or 35 ft 7 in. and a required lane width of 138 in. or 11½ ft.

Fig. 7 represents a tractor-semi-trailer with a 20-ft body, whose front wheels are turned to the same angles as in the previous example. Here we find a turning radius of but 319 in. or 26 ft 7 in. However, the lane width required for such a turn is 211 in. or 17 ft 7 in. Obviously, this would bring the rear of the trailer well over on the left side of the road.

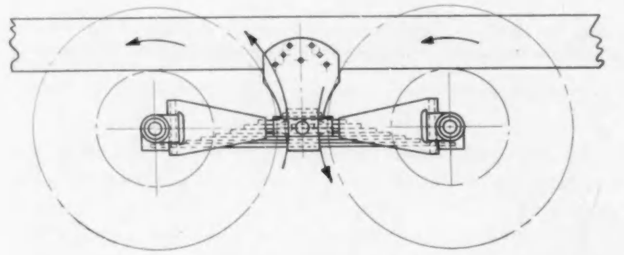
Fig. 8 shows the minimum radius at which the tractor-semi-trailer can turn within the same 11½-ft lane width



▪ Fig. 9 – Reach-member construction having a pronounced tendency to transfer weight from axle to axle, as well as a considerable torque moment on the frame



▪ Fig. 10 – Bogie suspension employing separate springs for each axle joined by evener bar at the center



▪ Fig. 11 – Radius-rod type of bogie suspension in which two radius rods pivoted close to the trunnion serve to position the axles and to transmit the brake torque

required by the six-wheeler. In this case the tractor-semi-trailer has a turning radius of 688 in. or 57 ft 4 in. – considerably more than the six-wheeler, so that no advantage can be taken of its sharper minimum turning radius without indulging in dangerous driving practices.

■ Bogie Hopping

Bogie hopping, as in the case of steering stability, is affected by weight transfer. In the latter we are concerned with the transfer between the bogie as a whole and the front axle, resulting from the inertia of the mass of the whole vehicle acting through the center of gravity. In the former, however, we must deal with forces acting within the bogie itself. These are concerned with inertia forces acting upon the two axles through the trunnion and, in addition, with the torque reactions of the axles.

There are two types of bogie construction on six-wheeled trucks, namely, the four-wheel-driven type and the two-wheel-driven type. In the former, it is customary to place the trunnion midway between the axles to effect even distribution of static weight between them. In the latter type, slightly more weight is imposed upon the driving axle than upon the trailing one – usually about 55% and 45%, respectively.

Obviously, for proportionate braking in the latter case, it is essential that slightly greater braking force be exerted on the brakes of the driving axle than on the trailing axle. In the case of the four-wheel-driven bogie, however, it would at first seem that the braking force applied to each axle should be the same.

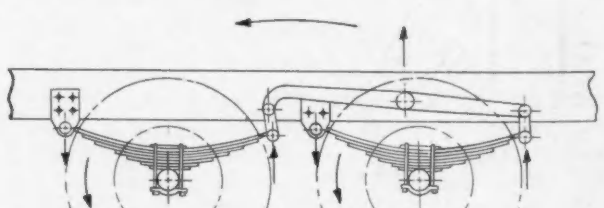
■ Torque Reaction

This is so, however, only if the dynamic loads remain balanced in the same manner as do the static loads. Unfortunately, in many types of bogie design, this condition is not realized. The reason is found in torque reactions from the axles to the frame and in the couple between the trunnion and the axle spindles, where the trunnion height is above the spindle height.

Bogie suspensions divide themselves into four general groups with respect to their torque reactions:

1. Those having a pronounced tendency to transfer weight from axle to axle as well as a considerable torque moment on the frame.
2. Those having a definite tendency to transfer weight between the axles but little or no torque moment on the frame.
3. Those having little or no tendency to transfer weight between axles, but still exert a torque moment to the frame.
4. Those having little or no tendency toward either.

In the first class is the reach-member construction illustrated in Fig. 9. This has a rigid reach member from axle to axle, to which the springs are clipped at their middles, their ends being shackled to the frame, thus forming which might be termed an elastic trunnion. The reach members take the torque of the axles, but receive no twist due to transverse vertical movement of the axles. This arrangement provides a wide load bearing on the frame and eradicates harmful twisting of the springs.



▪ Fig. 12 – Bogie suspension using separate springs for the two axles, the rear ends of which are joined by a long evener bar

The torque reactions of such a bogie will not only impart a strong torque moment to the frame, but will transfer considerable weight from the rearward to the forward axle.

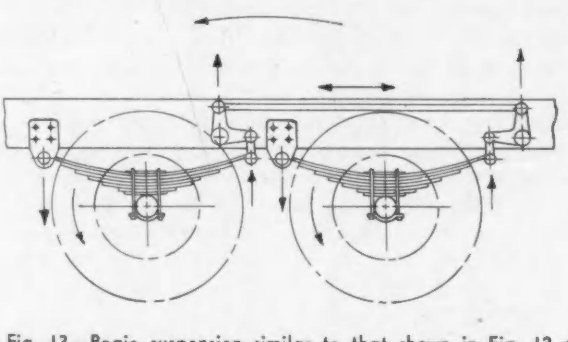
Another in this first class is that shown in Fig. 10. Here separate springs are used for each axle, trunnioned to the frame at their outer ends and joined by a short evener bar at the center. This arrangement is suitable only for use on relatively even road surfaces, as its flexibility is decidedly limited. On uneven surfaces there is danger of the evener bar turning too far, thus passing center and locking.

This arrangement reverses the usual order of weight transfer, actually transferring from the forward to the rearward axle. It also exerts a considerable torque moment to the frame.

In the second class is the radius-rod type, shown in Fig. 11, in which two radius rods, pivoted close to the trunnion serve to position the axles and to transmit the brake torque. This arrangement produces a strong transfer of weight, but a very slight torque moment on the frame.

One of two examples in the third class is that shown in Fig. 12. Here, as in Fig. 10, separate springs are used for the two axles, but the rear ends of each are joined by the long evener bar, thus eliminating axle-to-axle weight transfer. However, this suspension exhibits a definite torque reaction on the frame.

Fig. 13 depicts a similar arrangement except that through the use of bell-cranks, the long evener bar is replaced by a push-pull rod. Doubtless its designer fancied that this change eliminated the torque moment on the

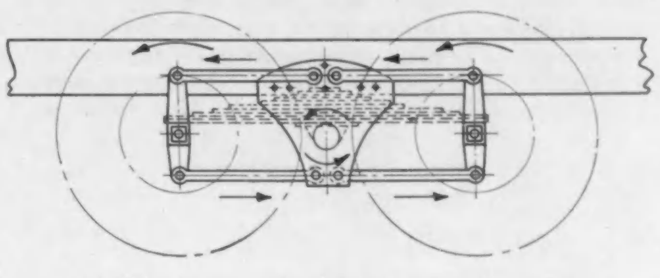


■ Fig. 13 – Bogie suspension similar to that shown in Fig. 12 except that, through the use of bell-cranks, the long evener bar is replaced by a push-pull rod

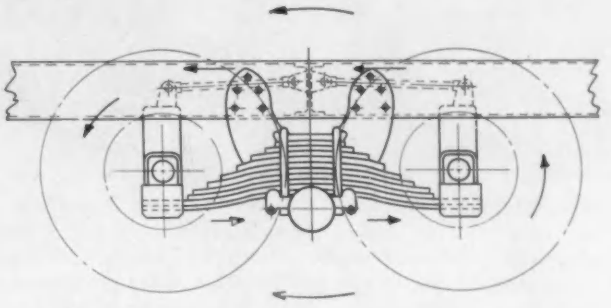
frame, but a moment's study will show that, as both bell-crank trunnions lift, it will act in exactly the same manner as the design just shown.

In the fourth class we have the familiar parallelogram system, illustrated in Fig. 14. Here, as the primary torque reaction of the axles is resolved into horizontal components, there is no weight transfer from axle to axle although there is some small degree of torque on the chassis frame. Although this design seems complicated, in practice it is somewhat simplified by using a single set of torque rods on top, located near the center of the frame.

Also in the fourth class is the construction shown in Fig. 15. This is a simplification of the parallelogram arrangement, in which the springs themselves serve as the



■ Fig. 14 – Bogie suspension employing parallelogram system



■ Fig. 15 – Simplification of the parallelogram arrangement of bogie suspension in which the springs themselves serve as the lower torque rods

lower torque rods, so that only the single upper set at the center of the frame are used. Twisting of the springs is eliminated entirely by the use of rubber shock insulators as the means of attachment of the springs to the axles.

It will be noted also that the trunnion is located below the level of the wheel spindles, so that the slight torque moment on the frame due to the torque rods is balanced by the opposing inertia force acting through the trunnion. This is the type of bogie assumed in the foregoing study of inertia weight transfer, in which all brake torque effect from the rear bogie was assumed to be absent.

■ Importance of Torque Reactions

Reverting to that study, it will be observed that the effect of brake torque upon load transfer is serious and present in both four-wheeled truck and four-wheeled tractor types. If the six-wheeler be equipped with a bogie which transfers any considerable amount of brake torque to the frame, then it, too, may overload the front axle well beyond its prudent braking capacity, thus reducing the overall brake effectiveness. However, since bogie constructions do exist which practically nullify torque reaction upon the frame, it will be evident that six-wheelers so equipped should, as they do, offer perhaps the closest approach to uniformly high, controllable brake ability.

■ Need for Stability

Obviously, regardless of weight transfer or the exactitude with which the braking distribution be adjusted to optimum conditions, neither the effectiveness of the brakes nor the directional control of the vehicle can be preserved

if the rear bogie is unstable. It is hoped that the foregoing examination of different bogie designs has clarified the matter sufficiently to demonstrate that bogie hopping is not an inherent fault of six-wheelers, but merely a manifestation of faulty design which can be overcome completely by employing a bogie which is torsionally balanced.

■ Effect of Inter-Axle Differential

Four-wheel-driven six-wheel bogies are made both with and without inter-axle differentials. It is not generally thought that this difference is of importance as affecting brakes, but a little reflection shows that, dependent upon the bogie construction, this difference may be important. Were there no differential action whatever, either between axles or between the wheels on each axle, then a single brake, acting anywhere on the system would have equal effectiveness to the conventional four-wheel brakes and unbalanced braking would be impossible. It would prevent the sliding of a wheel on a slippery spot unless all wheels slide at once and, in the case of a bogie suspension which failed to preserve equal distribution of weight between the axles, it would prevent the locking on the light axle, while the wheels on the heavy one continued to turn. However, in such case, the loads imposed upon the axle shafts would rise to four times normal while those on the driving gears and inter-axle driveshaft would be doubled. Other objections, of course, would be resistance to steering, inter-axle and inter-wheel fight, excess scuffing of tires and power-wasting frictional drag.

When only the inter-axle differential is omitted, quite different effects upon braking are produced. The locking of one wheel acts through the axle differential to speed up the opposite wheel. This acceleration is resisted by its brake and so a retarding force reacts through the solid inter-axle shaft upon the other axle. However, as both axles are constrained to turn at the same speed, a disproportionate amount of work will be done by the wheel opposite the slipping one. Indeed, this wheel may actually be accelerated slightly, thus reducing the total retarding force.

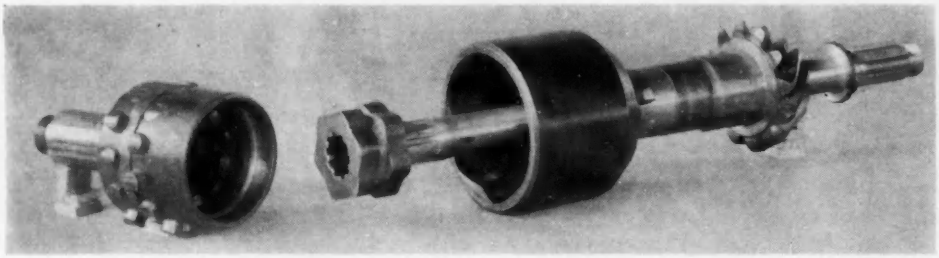
With three differentials, the loss of traction by any one wheel, upon a severe brake application, results in the locking of that wheel. This condition, in view of the forward travel of the vehicle, tends to speed up the opposite wheel, which acceleration is resisted by its brake, thus tending to slow down that axle. This slowing down of one axle, acting through the inter-axle differential, then, tends to speed up the other axle, thus increasing the work to be done by its brakes. Thus, to some extent, the loss of energy absorption on the wheel losing traction is made up for by increased friction on the other three.

Due to the action of the inter-axle differential, each axle may find its own balancing speed and hence the work done by the three effective brakes will be equalized with the minimum loss of total braking force.

■ Differential Types

One of the objections to third differentials, of course, regardless of their benefits to braking, is their effect upon driving traction. Straight differentials, such as universally used in axle carriers, have the unfortunate property of transmitting motion along the line of least traction resistance. This action is tolerable where but two wheels drive, but with four driving wheels, three such differentials make traction a bit uncertain.

This condition has given rise, during the past 30 years and more, to the invention and promotion of scores of self-locking or semi-self-locking differentials, ranging all of the way from simple ratchet devices up to adventures in unorthodox gearing. The great trouble in the past has been that, if the device were a true differential, it depended upon friction or inertia for its locking or semi-locking action. The former was wasteful of power, tended to become ineffective as the rough surfaces became polished and, like the latter, to develop destructive back-lash. Another type which has had considerable vogue employs a varying pitch line on an otherwise conventional-appearing bevel gear design to impart alternately favorable and unfavorable leverages to the two sides to gain considerable immunity to wheel-spinning. This type, in common with the inertia type, however, suffers the drawback of severe vibration – particularly when used in the high-speed inter-axle location and intermittent torque.



■ **Fig. 16 – Non-spinning differential which operates on a cam-and-plunger principle**

A non-spinning differential which provides a positive and limited torque bias toward the side having traction and wholly free from these objections, operates on a cam-and-plunger principle, providing true differential action at all times, with steady, continuous torque and no end-thrust whatever. This device, illustrated in Fig. 16, proves wholly adequate for highway work when used purely as an inter-axle power divider. For the severer classes of work it may be used in all three places.

It consists of inner and outer cams between which is a ring of radial plungers in two rows. The drive is applied to the central ring or cage, the inner and outer cams, respectively, forming the two sides of the differential. In straight-ahead driving, the plungers are balanced between the opposite slopes of the two cams, so that both are carried around with it. Upon rounding a turn, the faster side over-runs the central ring, its cams causing the plungers to shuttle in and out, reacting upon the opposite cams to retard their motion proportional to the increase in the

speed of the first cam. This provides perfect differential action, in which the average speed of the inner and outer cam members is always equal to that of the central ring.

This action takes place, however, only when one side is positively accelerated. Upon striking a slippery spot in the road, the cam connected with the wheel or axle affected cannot race, for the cam angles are such that even the small amount of resistance which it possesses reacts upon the plungers so as positively to turn the other side, whose cam is connected with the wheel or axle having traction. If the wheel be raised clear of the ground, as on a jack, of course, the resistance to its turning will be too small to produce this reaction, so that in this case it will spin idly. The action is such that the amount of torque thus biased in favor of the side upon which traction is preserved is always limited by the resistance of the side on which traction is lost, so that at no time can either side be subjected to more than 30% above its normal share of the torque.

This differential does not lose its effectiveness through polishing because its locking action is dependent upon cam angles and not upon sliding friction. It does not develop backlash because the combined areas of all of the plunger-to-cam contacts is far greater than in any set of differential gears and its torque cannot become irregular because the cam angles are perfectly complementary.

In conclusion, the authors wish to disclaim any antagonism toward the tractor-semi-trailer. If this paper has seemed to be a panegyric for the six-wheeler, it is because it was essentially defensive of that type. As may be seen in Appendix A, the majority of states discriminate against six-wheelers in fixing maximum weights. The preliminary report on "Brake Performance of Commercial Vehicles and Combinations" issued by the Bureau of Motor Carriers, ICC, March, 1941, included so small a sample of six-wheelers that no graphical report was made. The meager report given was decidedly unfavorable. In the preliminary report of the Public Roads Administration's studies of brake performance, the six-wheeler is not noticed at all. Consequently, being under attack from some quarters and pointedly ignored by public authority, the six-wheeler has seemed to be in desperate need of a champion.

Perhaps the authors have emulated Bill Stout's turtle in thus sticking their collaborative neck so far out of the protective shell of neutrality which commercial prudence would seem to recommend. Carrying the simile still farther, however, they wish to submit that, only by so doing, can a turtle or the profession of automotive industry make progress. Finally, it is the authors' profound conviction that the progress of the six-wheeler need not in any way impede the continued progress of the tractor-semi-trailer in its proper sphere of usefulness.

Appendix A
Legal Maximum Gross Vehicle Weights

State	Six-Wheeler	Tractor-Semi-Trailer	Favoring: Equal	Favoring: T-S-T
Alabama	30,000	30,000	X	
Arizona	34,000	40,000		X
Arkansas	44,800	53,900		X
California	51,200	73,600		X
Colorado	34,000	44,800		X
Connecticut	40,000	40,000	X	
Delaware	36,000	40,000		X
Dist. of Columbia	39,600	39,600	X	
Florida	24,000	40,000		X
Georgia	44,800	53,900		X
Idaho	42,000	42,000	X	
Illinois	40,000	40,000	X	
Indiana	45,500	50,400		X
Iowa	33,900	40,650		X
Kansas	34,000	46,900		X
Kentucky	18,000	18,000	X	
Louisiana	36,000	36,000	X	
Maine	40,000	40,000	X	
Maryland	58,400	65,250		X
Massachusetts	40,000	40,000	X	
Michigan	54,000	54,000	X	
Minnesota	42,000	54,000		X
Mississippi	30,000	30,000	X	
Missouri	24,000	38,000		X
Montana	44,800	54,000		X
Nebraska	32,000	40,000		X
Nevada	38,000	38,000	X	
New Hampshire	40,000	40,000	X	
New Jersey	40,000	60,000		X
New Mexico	38,400	46,200		X
New York	44,000	61,500		X
North Carolina	40,000	40,000	X	
North Dakota	35,000	40,000		X
Ohio	47,900	57,750		X
Oklahoma	24,000	47,000		X
Oregon	44,800	57,400		X
Pennsylvania	36,000	39,000		X
Rhode Island	40,000	40,000	X	
South Carolina	25,000	40,000		X
South Dakota	24,000	30,000		X
Tennessee	30,000	30,000	X	
Texas	38,000	38,000	X	
Utah	51,800	64,400		X
Vermont	40,000	40,000	X	
Virginia	35,000	35,000	X	
Washington	34,000	46,000		X
West Virginia	42,800	48,000		X
Wisconsin	36,000	43,000		X
Wyoming	41,400	46,200		X
Average	37,941	44,336	19	30

DISCUSSION

Supplements Discussion of Several Points

—L. R. Buckendale
Vice President in Charge of Engineering, The Timken-Detroit Axle Co.

MR. BENNING and Mr. Horine are to be congratulated on a very carefully prepared paper. However, there are a couple of points which I think should be cleared up.

The question of self-steering on a bogie is not new and involves some tricky problems if steering effect is obtained from a gravity factor. The problem can be worked out for all wheels on a plane surface but, when operating on a crowned road or on these high-speed banked highways at slow speed, the gravitational factor may react on the front wheels in the wrong direction.

Later in the authors' paper, the comment is made that a varying leverage type differential suffers the drawback of severe vibration. This statement is true when a wheel is spinning at high speed, but this is not a normal differential function. The differential should normally operate only at creeping speeds wherein the vibration factor becomes negligible.

On the subject of the discussion regarding whether the ground or the wheel spindle should be taken as the starting point of the forces, it is my opinion that the ground is the conservative thing because it gives a safer factor than the spindle, although the true point is probably somewhere between the ground and the spindle but, if this true point is to be determined, it will require a very careful analysis into all of the working forces because some of the forces, such as the brakes, become pretty tricky and care should be exercised that all of the polygons of forces be closed, or an erroneous answer will result.

The PRECISION of

Table 1 – Ranges Covered in Tests

Item	R-Samples: ASTM Motor Method min.	max.	CFR Res. Method min.	max.	A-Samples: ASTM Motor Method min.	max.
Octane Number of Fuels	52.9	90.4	55.3	98.7	73.1	1.22†
Greatest Deviation, octane units	+2.0	−2.2	−2.1*	+2.1*	−2.1	+1.8
Standard Deviation, octane number	0.17	0.92	0.13	0.89°	0.21	0.86
Barometric Pressure, in. hg	28.40	30.67	28.55	30.67	....	....
Humidity, grains per lb	8	145	9	178	....	....
Hours Run Since Carbon Removal	1	306	2	249	....	....
Compression Ratio	4.78	7.78	5.00	8.78	....	....
		min.	max.			
Sensitivity, octane units		+0.1	14.6			

† Tetraethyl lead in iso-octane, ml/gal.
* The values for R-199 were −6.2 and +3.9 respectively.
° The value for R-199 was 2.89.

Table 2 – Precision of R Sample Ratings Exchange Group – ASTM Motor Method

Laboratory Number	1939 Std. Dev.	1939 Const. Error	1940 Std. Dev.	1940 Const. Error	1941 Std. Dev.	1941 Const. Error	1939–1941 Std. Dev.	1939–1941 Const. Error
1	0.31	−0.09	0.23	+0.04	0.33	−0.10	0.30	−0.05
2	0.35	+0.20*	0.31	+0.03	0.26	+0.05	0.32	+0.10*
3	0.41	−0.09	0.41	−0.17	0.40	−0.09	0.41	−0.11*
4	0.51	+0.06	0.38	+0.12	0.41	+0.24*	0.44	+0.14*
5	0.35	−0.10	0.38	−0.12	0.34	+0.01	0.36	−0.07
6	0.49	−0.24*	0.48	−0.03	0.32	+0.12	0.46	−0.06
7	0.39	+0.06	0.26	−0.08	0.41	−0.29*	0.39	−0.10
8	0.55	−0.07	0.60	−0.21	0.58	−0.11	0.58	−0.13
9	0.46	−0.01	0.38	+0.08	0.51	+0.04	0.45	+0.03
10	0.44	+0.05	0.57	+0.01	0.49	−0.09	0.51	−0.01
11	0.63	−0.25	0.61	−0.22	0.55	−0.35*	0.60	−0.28*
12	0.48	−0.04	0.45	+0.15	0.46	+0.40*	0.50	+0.15*
13	0.52	+0.18*	0.49	+0.07	0.49	+0.21*	0.51	+0.14*
14	0.57	+0.26*	0.59	+0.19	0.51	+0.08	0.56	+0.18*
15	0.54	−0.19	0.45	−0.01	0.64	+0.08	0.56	−0.04
16	0.41	−0.10	0.57	+0.03	0.35	+0.12	0.47	+0.02
17	0.64	−0.12	0.39	0.00	0.47	+0.06	0.51	−0.02
18	0.66	+0.15	0.53	0.00	0.45	+0.06	0.56	+0.07
19	0.76	−0.13	0.60	−0.04	0.46	−0.38*	0.64	−0.17*
All	0.50		0.46		0.44		0.48	

* These values probably represent a real variation from the average.

Table 3 – Precision of R Sample Ratings Exchange Group – CFR Research Method

Laboratory Number	1939 Std. Dev.	1939 Const. Error	1940 Std. Dev.	1940 Const. Error	1941 Std. Dev.	1941 Const. Error	1939–1941 Std. Dev.	1939–1941 Const. Error
1	0.45	−0.15	0.38	+0.22*	0.29	+0.17*	0.38	+0.14*
2	0.52°	+0.02	0.41	+0.13	0.29	+0.01	0.38	+0.07
3	....	....	0.33	−0.03	0.36	−0.08	0.35	−0.05
4	....	....	....	....	....	....	....	....
5	0.54	+0.07	0.44	−0.14	0.38	−0.10	0.46	−0.06
6	0.60°	+0.05	0.29°	+0.12	0.28	+0.06	0.36	+0.06
7	0.35°	+0.02	....	....	0.55	−0.03	0.51	−0.02
8	....	....	....	....	....	....	....	....
9	0.49°	+0.43	0.61	−0.26	0.37	−0.03	0.50	−0.08
10	0.56°	−0.29	0.47	−0.09	0.41	−0.07	0.46	−0.10
11	....	....	....	....	....	....	....	....
12	0.57	−0.01	0.42	0.00	0.41	−0.15	0.49	−0.05
13	0.52	+0.27	0.39	−0.04	0.52	+0.04	0.48	+0.07
14	0.29°	−0.43	0.30	−0.15	0.38	+0.08	0.39	−0.02
15	0.62	+0.07	0.40	−0.05	0.39	+0.04	0.47	+0.02
16	0.54	−0.11	0.59	+0.26	0.47	−0.07	0.56	−0.04
17	0.44	−0.19	−0.40	+0.18	0.55	−0.04	0.50	−0.01
18	0.49	+0.22*	0.70	−0.06	0.54	+0.12	0.57	+0.12
19	0.51	−0.60*	0.59	−0.39	0.78	+0.08	0.69	−0.32*
All	0.50		0.45		0.44		0.47	

° Less than 10 values.
* These values probably represent a real deviation from the average.

I. Introduction

IN 1934, the Cooperative Fuel Research Committee decided that sufficient knock-rating data were at hand, as a result of cooperative testing of Exchange Group and other samples, to warrant an analysis with the object of determining the normal precision of rating and the factors affecting precision. The National Bureau of Standards undertook this work, and carried out an exhaustive analysis[1] of 2180 cooperative tests made on 99 fuels including the first 67 R-samples, thus covering the year 1935 as well. Three years later, a second analysis[2] was made, covering 6386 tests on 136 fuel samples rated in 1936-1938.

The first two analyses established rather definitely the factors which affect precision of rating. These were found to include atmospheric humidity, engine carbon, knock intensity, and possibly in certain cases, air temperature. A limit was subsequently set on the time of operation between overhauls to remove carbon, and work on means for humidity control was begun. Since the last analysis, limits have been set on humidity and engine air temperature, and means have been developed for determining more accurately the standard knock intensity. Thus all factors known to affect precision are now controlled or limited. Consequently, only a minor portion of the present work has been devoted to the study of their effect.

The present analysis comprises 6925 tests made on 183 fuel samples during 1939-1941. It covers ratings on 109 R-samples, including tests by non-member participants on 19 semi-annual samples, on 73 A-samples, and on the sensitive reference fuel X-1. The ranges covered in these tests are given in Table 1.

The statistical treatments used in this analysis follow the methods of Fisher,[3] Ezekiel,[4] and Simon.[5]

II. Results in Brief

In the period covered by this analysis the standard deviation of Exchange Group ratings has averaged 0.48 octane unit by the ASTM Motor Method and 0.49 octane unit by the CFR Research Method. Since the last analysis, no improvement has been shown in the rating of straight-run fuels, and the greatest improvement shown is in the rating of clear and leaded 100% cracked fuels, with lesser improvement in the rating of blends of cracked and straight-run fuels, both clear and leaded.

Of the 2291 ASTM Motor Method ratings made on R-samples by the Exchange Group, 92.9% showed deviations of less than 1 octane unit, and only 0.1% showed deviations of 2 octane units or more. Of 1258 CFR Research Method tests on R-samples, excluding the maverick R-199, 94% showed deviations of less than 1 octane unit. The best Exchange Group laboratory had an average standard deviation for ASTM Motor Method ratings of 0.30 octane unit for the three years covered by this analysis

MOTOR FUEL TESTING

Report from Cooperative Fuel Research Committee

by DONALD B. BROOKS* and ROBETTA B. CLEATON*

THIS third triennial analysis of the precision of motor fuel testing is based on 6925 knock ratings on 183 fuels, and inspection data on 109 of these fuels. The analysis shows that the precision of rating by both ASTM Motor and CFR Research Methods has continued to improve during the last three years, the standard deviations for both methods now being below 1/2 octane unit. It appears possible for these deviations to be reduced to half of this value.

The humidity control apparatus is shown to have no direct effect on ratings. For the first time, the analysis includes fuel inspection data, and estimates of precision are given for vapor pressure, gravity, and distillation measurements.

■ ■ ■

THE AUTHORS: DONALD B. BROOKS (SM '24) tells us that he first developed a liking for things automotive in 1918 when he was in the 332nd Battalion, Tank Corps, U. S. Army. Senior automotive engineer with the National Bureau of Standards, Washington, D. C., prior to 1927 Mr. Brooks earlier had been with the Studebaker Corp., The Texas Co., and in other capacities at the Bureau of Standards. Ohio State University granted him his B.Ch.E., M.Sc., and Ch.E. degrees. MRS. ROBETTA B. CLEATON, one of the few women to contribute material published in the SAE Journal, has been engaged in statistical research connected with motor fuel problems, and in research on the impurities present in normal heptane and iso-octane, the primary reference standards of knock testing, since joining the National Bureau of Standards in 1936. She was born in Scotland and educated in England and America. Before joining the Bureau of Standards she did statistical work for the U. S. Bureau of Census.

and, during one of these three years, it averaged 0.23 octane unit. By the CFR Research Method, the lowest average standard deviation for the three years was 0.35 octane unit, but one laboratory averaged 0.28 octane unit for two consecutive years. These figures, in the light of present trends by both methods, suggest that the ultimate precision by each method, with present equipment, is close to 0.2 octane unit.

The standard deviations of ASTM Motor Method ratings increase with increase in sensitivity of the fuel, while those of CFR Research Method ratings decrease with increase in sensitivity. Another curious, and probably related, dissimilarity in results by the two methods is the fact that, by the Motor Method, straight-run fuels are rated with the best precision and 100% cracked fuels with the least while, by the Research Method, almost the opposite is true.

By both methods, leaded straight-run fuels are rated less precisely, and leaded 100% cracked fuels more precisely, than the corresponding clear fuels.

An average curve relating compression ratio and octane number, which was derived in the 1939 analysis,[2] is shown to be practically identical, below 90 octane number, with the recently adopted guide curve[6] for the ASTM Motor Method.

The expected effects of humidity and engine carbon are apparent in the ratings of some fuels, notably cracked gasolines. The humidity control apparatus does not appear *per se* to affect ratings. The sensitive fuel X-1 deteriorated after the winter of 1941, the loss of knock rating being larger by Motor than by Research Method.

Standard deviations are derived for the fuel inspection measurements. Reid vapor pressure is shown to be correlated closely with the initial boiling point. Loss can be estimated more accurately from the initial boiling point than by indirect measurement.

III. Precision of Rating

Tables 2 and 3 list the precision of rating by laboratories

* National Bureau of Standards.

[1] See SAE Journal, Vol. 39, October, 1936, pp. 22-24: "The Precision of Knock Rating," by Donald B. Brooks.

[2] See SAE Transactions, October, 1939, pp. 449-456: "The Precision of Knock Rating – 1936-1938," by Donald B. Brooks and Robetta B. Cleaton.

[3] See "Statistical Methods for Research Workers," by R. A. Fisher, Oliver & Boyd, Ltd., Edinburgh.

[4] See "Methods of Correlation Analysis," by Mordecai Ezekiel, John Wiley & Sons, New York, N. Y.

[5] See "Engineers' Manual of Statistical Methods," by Leslie E. Simon, John Wiley & Sons, New York, N. Y.

[6] See *Journal of Research of the National Bureau of Standards*, Vol. 28, No. 6, June, 1942, pp. 713-734: "The Effect of Altitude on Knock Rating in CFR Engines," RP 1475, by Donald B. Brooks.

for the ASTM Motor and CFR Research Methods. Data are given by years as well as for the three-year period.

The standard deviation given for each laboratory is corrected for the constant error, given in the next subsequent column. This procedure is equivalent to computing the standard deviation of each laboratory about its own mean value. The mean amount by which tests of a laboratory differ from the average value found by the participating laboratories is called the constant error.

As measurements are always accompanied by errors, it is obvious that accidental errors may give rise to a residual which cannot be differentiated from a constant error. For any particular case, it is possible to set a limit to the size of constant error which is likely to so accrue. Values in Tables 2 and 3 which are highly unlikely to have resulted in this manner are marked with an asterisk. For example, in Table 2 the constant error of —0.28 octane unit, shown by laboratory No. 11 for 1939-1941, would result from the accumulation of chance errors only once in 100,000 times.

In preparing Table 3, the CFR Research Method ratings on sample R-199 have been discarded. This cracked fuel of low front-end volatility resulted in the largest errors ever recorded in Exchange Group testing. The standard deviation of rating this fuel was 2.89 octane units; two of

Table 4 – Precision of Rating Semi-Annual Samples

Group	ASTM Motor Method No. of Labs.	ASTM Motor Method Std. Dev. Oct. Units	ASTM Motor Method % of 1939 Std. Dev.	CFR Research Method No. of Labs.	CFR Research Method Std. Dev. Oct. Units
Non-Member Participants					
Oil Companies	69	0.711	107	34	0.674
State Laboratories	18	0.741*	88	1	1.415
Commercial and Miscellaneous	13	0.585	93	4	0.873
All Non-Member	100	0.703°	101	39	0.707
CFR Exchange Group	19	0.482	99	16	0.501
All Participants	119	0.662°	102	55	0.631
	ASTM Motor Method			CFR Research Method	
Total Number of Ratings	1781			555	

* With 2 ratings omitted. Inclusion of these values gives 1.378.
° With 2 ratings omitted.

the deviations exceeded 5 octane units; and 6 of the 13 values differed from the mean by 2 octane units or more. As inclusion of the ratings on this sample would increase the variance for all Research ratings on the 109 R-samples by 50%, it has been treated as a special case.

No cause is assignable for the anomalous behavior of R-199. Its standard deviation by the ASTM Motor Method is 0.59 octane unit, an entirely normal figure; yet, by the CFR Research Method, the standard deviation is 2.00 octane units higher than that for any other R-sample. No ratings can be rejected on the "gross-error" principle; that is, the *distribution* of errors is not abnormal. It is necessary – and statistically unjustifiable – to reject 5 of the 13 ratings to reduce the standard deviation to a figure below that for the next worst sample. No significant correlation is shown between the rating obtained and the atmospheric humidity or the carbon deposition (expressed as engine-hours since cleaning). Because of the relatively enormous errors involved in rating this sample, it is suggested that this fuel, if reproducible, should be considered as an engine condition standard for the CFR Research Method. Some correlation is shown between the error of rating R-199 and the standard deviation for the rest of the R-samples.

It will be noted from Tables 2 and 3 that there has been a consistent slight improvement in the precision of knock rating by the ASTM Motor Method, and a consistent

Table 5 – Deviations of Ratings of R-Samples

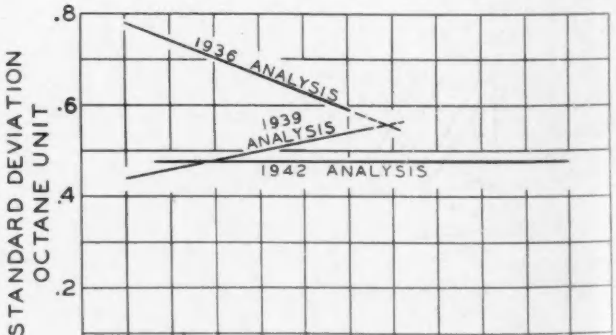
Year of Analysis	Test Method	No. of Tests	Std. Dev.	Percent of Total Number of Deviations in Range 0.0-0.9	1.0-1.9	2.0-2.9	3.0-Over
1936	ASTM Motor	1882	0.63	86.9	11.9	1.1	0.1
1939	ASTM Motor	2493	0.52	92.5	7.3	0.2	0.04
1942	ASTM Motor	2291	0.48	92.9	6.9	0.1	0.0
1942	CFR Research*	1258	0.49	94.0	5.6	0.2	0.08
1942	CFR Research	1271	0.56	93.4	5.8	0.5	0.3

* Excluding R-199

marked improvement in the CFR Research Method precision. The occurrence of real constant error is much less frequent in the case of the latter method. It is probable that the cause of the real constant errors lies in individual technique in obtaining bouncing-pin adjustment, rather than in inherent differences in engines, and this probability is supported by the fact that this adjustment is distinctly less critical in the Research Method.

In passing, it is desired to express admiration for the work of the laboratory listed as No. 1 in Tables 2 and 3. In the first of these analyses, their precision was within 6% of what was then calculated to be the maximum precision attainable with equipment and technique then available. In the second analysis, it was within 10%. Throughout the nine years of the CFR Exchange Group's work, this laboratory has averaged highest in precision, and for many years has not had an ASTM rating in error by as much as 1 octane unit. This is an excellent example of what *can* be done.

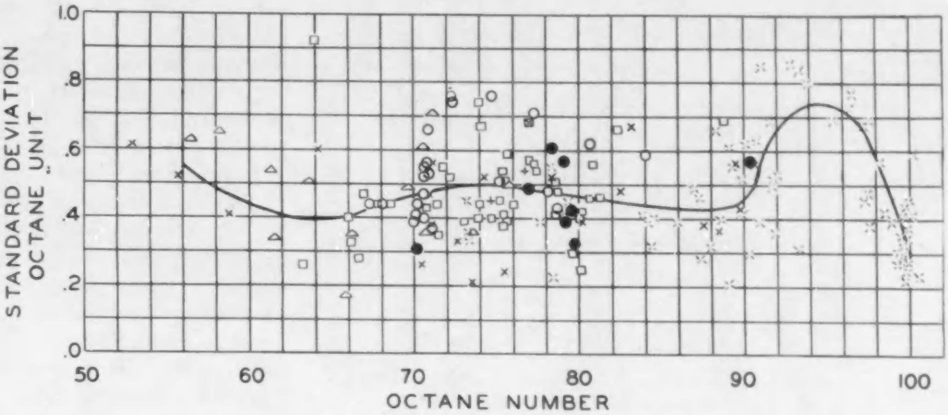
A special analysis was made of all ratings on the Semi-Annual Cooperative test fuels to determine the precision by classes of the laboratories involved. A total of 2336



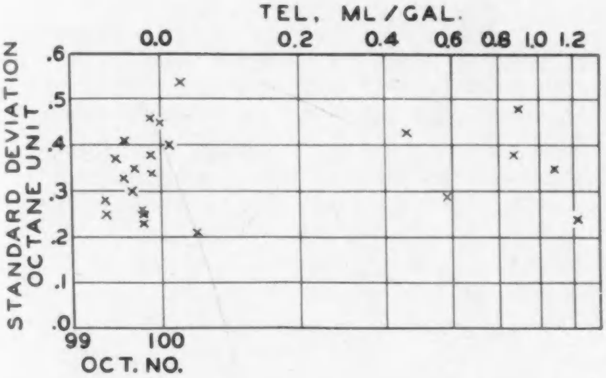
■ Fig. 1 – Precision of rating by ASTM Motor Method as shown by 1936, 1939, and 1942 analyses

■ Fig. 2 – Variation of precision of rating with octane number – ASTM Motor Method

Crosses are straight-run fuels, clear or leaded; open crosses, aviation fuels; open circles, clear 100% cracked fuels; solid circles, leaded 100% cracked fuels; triangles, clear blends of straight-run and cracked; squares, leaded blends of straight-run and cracked; and plus signs, benzol blends



ratings was reported on the 19 fuels. Table 4 gives the standard deviation of each class of laboratory for both test methods, and compares the standard deviation by the ASTM Motor Method with the results shown in the last periodic analysis. State and commercial laboratories have shown an improvement in the three-year interval, but the precision of rating by non-member oil companies has deteriorated. As in the previous analysis, the higher precision of the Exchange Group ratings demonstrates the value of regular participation in exchange sample testing.



■ Fig. 3 – Precision of ratings near 100 octane number, in octane units – ASTM Motor Method

The breakdown of deviations into ranges is given in Table 5 for both methods. For comparison, results reported in the two previous analyses are included. The improvement in the ASTM Motor Method ratings, noted in 1939, has continued although to a lesser degree. In the past six years of Exchange Group testing, covering nearly 5000 ratings, only one rating has deviated from the mean by 3 octane units. Tests by the CFR Research Method show a higher percentage of ratings falling within 1 octane unit of the mean, but also a higher percentage deviating by 2 or more units. This condition probably results from the newness of the method. Within the past year, the precision of Research ratings has equaled that of Motor ratings, and this trend may be expected to continue.

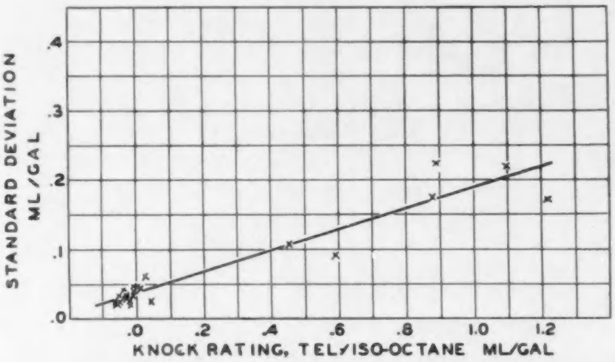
IV. Variation of Precision with Octane Number

In studying the variation of precision with octane number, 972 ASTM Motor Method tests on 73 (aviation) A-samples were included. Totally as well as separately, the 3263 tests on 182 fuels show no uniform tendency toward increase or decrease of precision with increase in octane number of the fuel tested. The comparison of this general result with those obtained in former analyses is shown in Fig. 1. From this figure it is obvious that most of the improvement in precision shown in the present analysis results from reduction of error in the higher octane range. It appears probable that this resulted from the establishing of a guide point at 90 octane number.

Fig. 2 is a detailed plot of the standard deviation for each sample against the octane number of the sample. The curve in Fig. 2 is drawn through averaged values for successive ranges of octane number. A minimum is shown at 65 octane number, which was a guide point, and at 88 octane number, near the former 90 octane number guide point. Beyond that point the error of rating rises rapidly to a maximum near 94 octane number, then falls sharply and, at 100 octane number, reaches the lowest point of the entire curve, a standard deviation of 0.30 octane unit.

The explanation for these peculiarities may be as follows: The bouncing pin was customarily set at 65 and at 90 octane number, and was used somewhat beyond these ranges. In recent work on the guide curve, it was found that, while a setting could readily be obtained which was satisfactory from 40 to 90 octane number, the same setting in no case was satisfactory much above the latter figure. It is possible that pins set at 65 and 90 octane number often were used to rate fuels up to 93 or 94 octane number, whereas a resetting would have been advisable. However, a pin set at 65 and 90 octane number would, in general, be unusable and would demand resetting above 95 octane number. The explanation of the high point in Fig. 2 near 94 octane number would thus seem to lie in the use of a pin not adjusted for this range. If this be true, an improvement in this range should be shown hereafter, in consequence of using a continuous guide curve in place of the former two fixed points.

The ASTM Motor Method ratings of certain samples provide means for determining whether the standard deviation varies materially above 100 octane number. Fig. 3 shows the results found for samples rating above F-3. The standard deviations of samples rating above iso-octane have been converted to octane units by an indirect method based on the variation of compression ratio with octane number below 100. It will be seen that there is no marked trend toward higher standard deviations above 100 octane number, and that these deviations average below the mean value found from tests at all octane numbers.



■ Fig. 4 – Precision of ratings near 100 octane number, in ml of $PbEt_4$ per gal of iso-octane – ASTM Motor Method

When expressed in terms of milliliters of tetraethyl lead per gallon of iso-octane, Fig. 4, the standard deviations show a steady increase, owing to the diminishing effect of higher concentrations of tetraethyl lead.

Analyses of the CFR Research Method data were made on all 109 R-samples, and on 108 R-samples with the anomalous fuel R-199 excluded. In the absence of the latter, a definite correlation was shown between the standard deviation and the octane number of the fuel, the deviation decreasing with increase of octane number. This correlation, which has a random probability of 1 in 1000, indicates an average standard deviation of 0.58 octane unit at 60 octane number and 0.32 at 100 octane number. The trend toward lower deviations at high octane number is apparent from the plot of these data, Fig. 5.

V. Variation of Precision with Sensitivity

A study of the deviations of ASTM Motor Method ratings in comparison with the difference in rating of each fuel by Motor and Research Methods ("sensitivity") shows a small but rather definite correlation, the deviation tending to increase with increase in fuel sensitivity. A fuel of zero sensitivity, it is indicated, would, on the average, be rated with a standard deviation of 0.44 octane unit, while one having a sensitivity of 10 octane units would result in a standard deviation of 0.53 octane unit. The data on which these deductions are based are plotted in Fig. 6. While the data may seem to scatter with little or no apparent trend, correlation shows that the tendency for the standard deviation to increase with increase in fuel sensitivity is such as would occur by chance only once in 75 such sets of data. Stated otherwise, the random probability of as high a correlation as that found is 1/75.

The CFR Research Method data also were analyzed. In this case the random probability was 1 in 200, indicating more definite relation of standard deviation to fuel sensitivity. However, the data, shown in Fig. 7, indicate that the standard deviation in this case decreases with increase in sensitivity. Both of the foregoing statements are based on the 108 R-samples, not including the anomalous R-199. Inclusion of the latter annuls the correlation and reverses the conclusion, as this fuel of enormous standard deviation also had the highest sensitivity, as can be seen in Fig. 7. It is believed, however, that the conclusions drawn from tests on the other 108 R-samples should be considered valid.

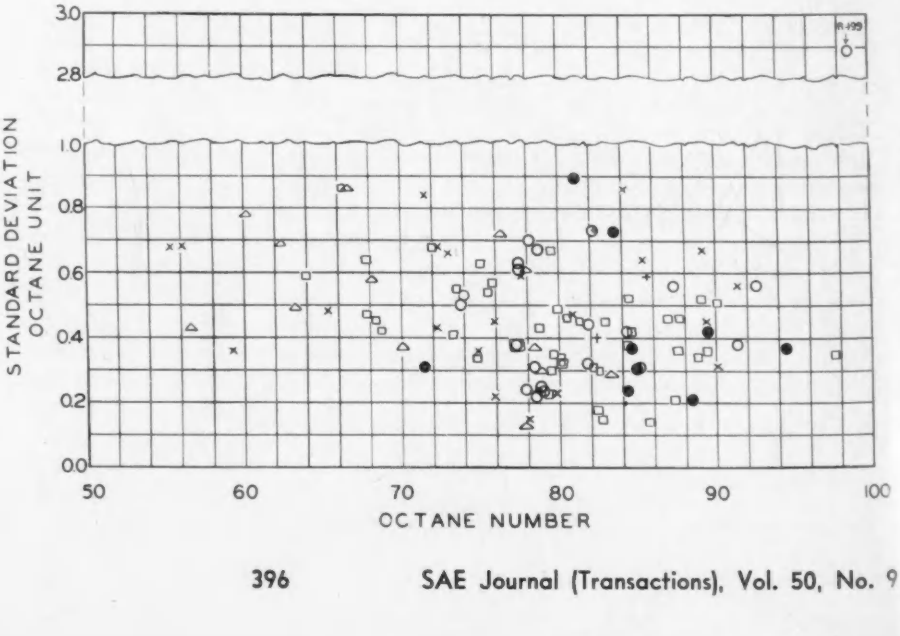
VI. Relation of Precision to Fuel Type

The relation of precision of rating to fuel type is shown in Table 6 for both test methods. Similar information derived in past analyses[1, 2] is included in this table for comparison. In the earlier analyses, however, clear and leaded straight-run fuels were not segregated, as the first analysis showed the precision of straight-run fuels to be unaffected by addition of tetraethyl lead.

While the ASTM Motor Method data show that no improvement has occurred in the precision of rating for combined clear and leaded straight-run fuels, clear straight-run fuels are rated more precisely than any other type. Since the 1936 analysis, improvement is shown in the rating of all other types of fuels, and is most pronounced in the case of clear 100% cracked fuels. It seems probable that this marked improvement has resulted at least in part from the general use of the sensitive reference fuel X-1 as a check on engine condition during much of the period covered by the present analysis. For comparison, the precision of rating the aviation exchange A-samples is included in this table. The A-samples are

■ Fig. 5 – Variation of precision of rating with octane number – CFR Research Method

Crosses are straight-run fuels, clear or leaded; open crosses, aviation fuels; open circles, clear 100% cracked fuels; solid circles, leaded 100% cracked fuels; triangles, clear blends of straight-run and cracked; squares, leaded blends of straight-run and cracked; and plus signs, benzol blends



rated somewhat more precisely than the R-samples in general.

In considering the CFR Research Method ratings, the data on sample R-199 have been excluded, as this sample is definitely a special case. In contrast to the ASTM Motor Method, the CFR Research Method is least precise in the rating of straight-run fuels as a class, although segregation of clear and leaded samples shows this condition to result largely from the relatively large errors involved in rating leaded straight-run fuels by this method. Precision above the general average is shown in the rating of cracked fuels, both clear and leaded, and of leaded blends of straight-run and cracked fuels.

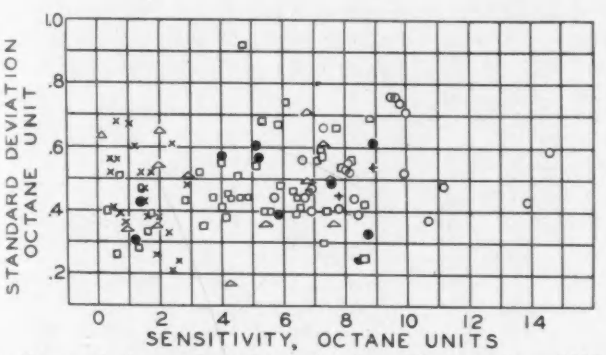
By both methods, leaded 100% cracked fuels are rated more precisely than clear 100% cracked fuels, and leaded straight-run fuels much less precisely than clear straight-run fuels.

VII. Ratings of Sensitive Reference Fuel X-1

The sensitive reference fuel X-1 has been rated by the ASTM Motor and the CFR Research Methods more or less regularly since July, 1940. It has also been rated as an unknown four times, under the designations R-202, R-212, R-258, and R-260. These ratings and their standard deviations, plotted in Figs. 8 and 9, reveal several interesting points.

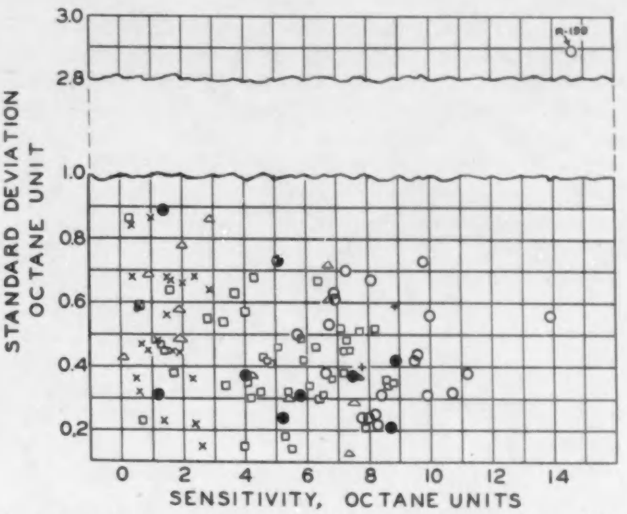
The dashed and solid lines on these figures delimit the zones of fairly high and very high probability, and are based on the first twelve months' testing of X-1 as a *known* sample. The variation of these limits is occasioned by changes in the number of ratings on which the monthly averages, plotted herein, are based. For simplicity, minor variations in this number have been disregarded. As the methods of test were not changed in the entire course of these tests, the departure of either statistic (average rating and standard deviation) from the inner zone makes it likely, and departure from the outer zone makes it almost certain, that something else changed.

Considering first the ASTM Motor Method ratings, Fig. 8, it appears that, both as an unknown and as a known sample, this fuel appeared to behave normally through May, 1941, although the March, 1941, average rating



■ **Fig. 6 – Variation of precision of rating with sensitivity – ASTM Motor Method**

Crosses are straight-run fuels, clear or leaded; open crosses, aviation fuels; open circles, clear 100% cracked fuels; solid circles, leaded 100% cracked fuels; triangles, clear blends of straight-run and cracked; squares, leaded blends of straight-run and cracked; and plus signs, benzol blends



■ **Fig. 7 – Variation of precision of rating with sensitivity – CFR Research Method**

Crosses are straight-run fuels, clear or leaded; open crosses, aviation fuels; open circles, clear 100% cracked fuels; solid circles, leaded 100% cracked fuels; triangles, clear blends of straight-run and cracked; squares, leaded blends of straight-run and cracked; and plus signs, benzol blends

indicated the shape of things to come, and the initial unknown ratings suggest that the reference fuel knock rating assigned to it was somewhat too high. After May, 1941, however, no amount of effort could keep its rating up, and subsequent values clearly indicate its degradation. Finally, in November and December, 1941, it was rated as an unknown, and its rating was shown to have fallen to 70 octane number.

The upper portion of Fig. 8 shows the behavior of the standard deviation during this period. In every case in which X-1 was rated as a known, the standard deviation of rating is within normal limits. This condition results from the fact that it was ordinarily used to standardize engines, and efforts were made to adjust the test apparatus so as to obtain a certain rating for it. The fact that the standard deviations remained within the normal limits after degradation of its rating set in is evidence that the

Table 6 – Relation of Precision to Fuel Type
Standard Deviation of Exchange Group Ratings

	ASTM Motor Method						CFR Research Method	
	1936 Analysis		1939 Analysis		1942 Analysis			
Type of Fuel	Oct. Units	% of Avg.	Oct. Units	% of Avg.	Oct. Units	% of Avg.	Oct. Units	% of Avg.
Straight-Run, Clear	0.60	95	0.46	87	0.41	84	0.44	97
Straight-Run, Leaded					0.52	107	0.59	129
Clear Blend of Straight-Run and Cracked	0.63	100	0.51	97	0.48	99	0.51	111
Leaded Blend of Straight-Run and Cracked	0.59	93	0.51	98	0.48	98	0.42	92
Clear 100% Cracked	0.74	118	0.61	116	0.53	110	0.45*	98
Leaded 100% Cracked	....	...	0.55	106	0.49	101	0.44	96
Benzol Blends	....	...	....	...	0.50	102	0.50	108
All R-Samples	0.63	100	0.52	100	0.48	100	0.46*	100
A-Samples	....	...	....	...	0.46	95	....	...

* Excluding R-199. With R-199 included, clear cracked averages 0.57, and all R-samples, 0.48 octane unit.

extent of its deterioration was generally recognized. When it was rated as an unknown, however, no effort was made to secure a specific value, and the standard deviations, all of which are outside the "fairly probable" limits for known ratings, average 0.475, which is very close to the average for all R-samples.

The CFR Research Method ratings, Fig. 9, differ chiefly in that the degradation of rating is much less. Where standard deviations are omitted, less than four ratings were reported. The ratings and the standard deviations for two of the four "unknown" sets of tests are outside the zones of "very high" probability, and indicate that some effort was made to secure an expected value for the rating when tested as a known sample. This inference is supported by the rather definite break between the "known" and "unknown" ratings found for it in November, and December, 1941.

It can be concluded that the initial values assigned to X-1 were too high by about ¼ octane unit by both methods; that its rating decreased, especially during the latter part of 1941, and decreased more by the ASTM Motor Method than by the CFR Research Method.

VIII. Effect of Humidity Control Apparatus on Ratings

Ratings with humidity controlled by standard or nonstandard ice towers have been made on most of the R-samples covered by this analysis. In many instances, particularly in the case of the CFR Research Method, the observed data were too few to establish a really reliable value of the mean rating with controlled humidity. However, the tendencies shown by the entire groups of Motor and Research ratings are consistent, and statistical tests indicate that reliance may be placed on them.

The ASTM Motor Method ratings at a humidity of 27 grains per lb average 0.2 octane unit below the ratings made without humidity control. The controlled humidity ratings on the 7 samples whose ratings at uncontrolled humidity showed definite correlation with humidity, averaged 0.6 octane unit below the ratings of these samples at uncontrolled humidity. It can consequently be inferred that the general average difference of 0.2 octane unit just noted can be ascribed to the effect of humidity, and not to the effect of the control apparatus. To account for the observed difference, the average response need be a decrease of only 1/3 octane unit per 100 grains per lb increase of humidity.

Although fewer ratings by the CFR Research Method are available, the indications are analogous. It can be concluded with reasonable certainty that use of the standard ice tower, or apparatus of similar design, does not affect ratings by either method.

IX. Correlations

The 1936 and 1939 analyses[1, 2] established definitely the correlation of compression ratio for standard knock intensity with barometric or dry-air pressure and with hours run since carbon removal. These analyses also established the correlation of octane number with compression ratio and with hours run since carbon removal and, in the case of some types of fuels, with atmospheric humidity.

Because the former analyses appeared conclusive in regard to these correlations, and in view of the large amount of labor which would have been required, no such extensive studies were made in this analysis. By inspection, fuel rating data indicating some correlation between octane number and humidity or hours since carbon removal were selected for further analysis.

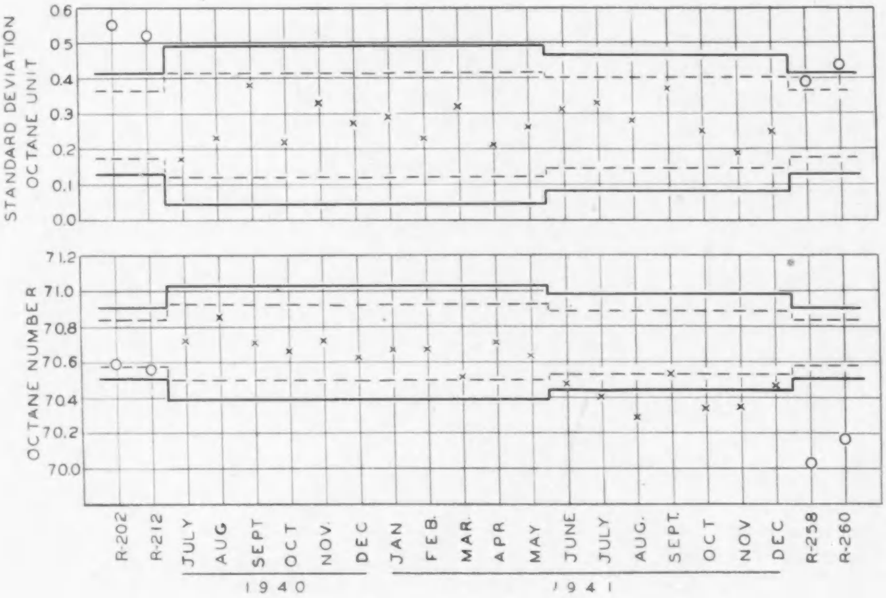


Fig. 8 – Ratings and standard deviations of Fuel X-1 – ASTM Motor Method

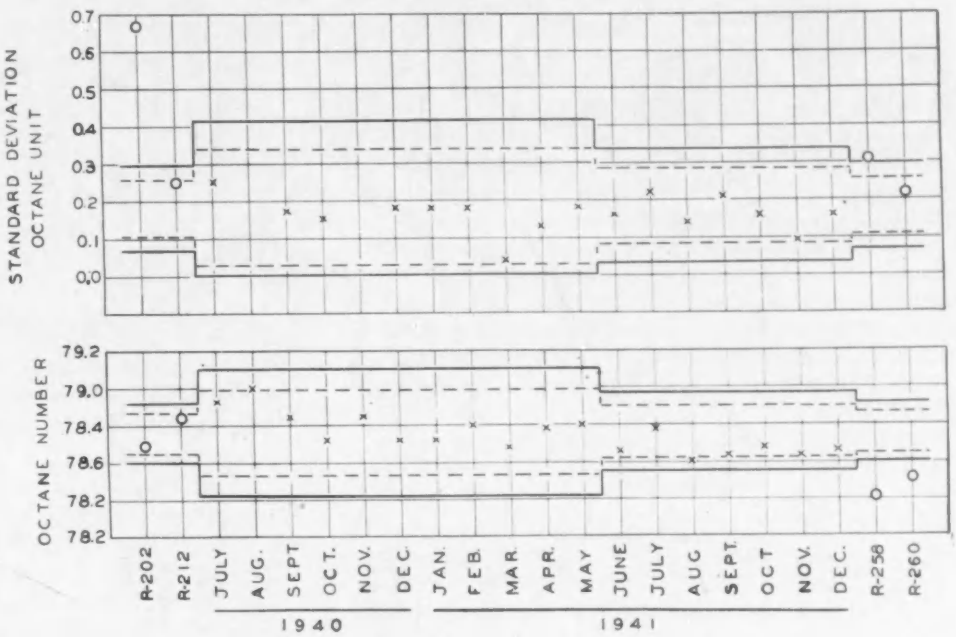
The crosses represent ratings as a known fuel, and the circles as an unknown fuel. Departure of either statistic from the inner zone represents probable, and from the outer zone, certain change in conditions surrounding the determinations

Definite correlations between ASTM Motor Method knock ratings and humidity were found in the case of 9 R-samples, each of which showed an increase of rating with increase of humidity. The amounts of increase varied from 1 to 2 octane units per 100 grains per lb change of humidity, and averaged 1.4 octane units. For the different samples, the random probability of the correlations varied from 1/20 to 10^{-8}.

Only 6 of the R-samples showed definite correlation between CFR Research Method ratings and humidity. In two of these cases the data indicated a decrease of rating

of slightly over 2 octane units per 100 grains per lb increase of humidity. In the other 4 cases, the increase of rating per 100 grains per lb increase of humidity varied from 0.8 to 2.2 octane units, and averaged 1.4 octane units. These latter 4 cases also showed correlation by the ASTM Motor Method, but the first two did not.

Of the 11 samples in which knock rating by either method was found to be correlated with humidity, 10 contained cracked gasoline, and 9 of the 11 samples contained tetraethyl lead. These figures are definitely in excess of the proportion of cracked and leaded fuels to the total number of R-samples, and substantiate the observations made in the previous analyses: that the greatest response to humidity was shown by leaded cracked fuels.



■ **Fig. 9 – Ratings and standard deviations of Fuel X-1 – CFR Research Method**

The crosses represent ratings as a known fuel, and the circles as an unknown fuel. Departure of either statistic from the inner zone represents probable, and from the outer zone, certain change in conditions surrounding the determinations

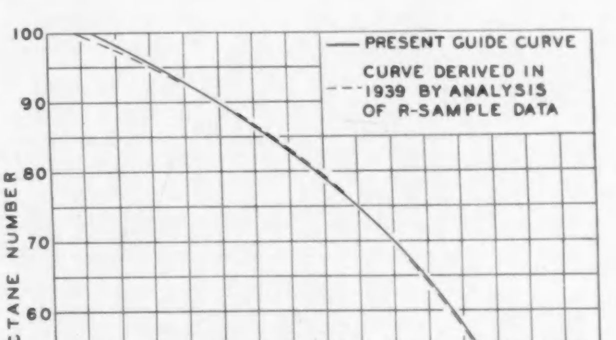
Correlations also were determined on selected cases between knock rating and hours since carbon removal. In all cases the correlation obtained was slight. Both earlier analyses showed definite correlation between these variables. It is probable that the lower degree of correlation shown in the present work is the result of general adoption of shorter overhaul periods, the effect of engine carbon thereby being reduced to the point where it is masked by the operation of other sources of error.

Correlations made on the anomalous sample R-199 showed less than 1 octane unit increase per 100 grains per lb increase of humidity in rating by the ASTM Motor Method, and over 8 octane units by the CFR Research Method. In both cases the random probability, however, was about 1 in 4, so these correlations can be considered very doubtful. The correlations between rating and hours since carbon removal were even more uncertain. The observed octane number by the CFR Research Method showed a very high correlation with the compression ratio used for testing. If this fuel can be duplicated, further study of it would be worth while.

X. Guide Curve for the ASTM Motor Method

In the 1939 analysis[2], the experimental data on 51 R-samples, on the calibration of reference fuel A-5, and on the certification of several batches of n-heptane and iso-octane, were analyzed to determine the relation between compression ratio and octane number. The results were expressed as an equation relating octane number and micrometer setting, and it was noted that the average values for 114 samples of R-, S-, and W-fuels were well fitted by the equation.

In view of the fact that the foregoing equation was derived statistically from data obtained in the routine rating of samples by all Exchange Group members, it is of interest to compare it with the guide curve[6] for the ASTM Motor Method which was determined by tests made specifically for the purpose. In Fig. 10, the dashed line shows



■ **Fig. 10 – Comparison of present guide curve for the ASTM Motor Method with average curve deduced from tests made in 1936, 1937, and 1938**

the relation derived in 1939 and expressed by the equation:

$$x = 101.465 - 41.589y + 6.411y^2 - 98.965y^3 \quad (1)$$

where
y = micrometer setting, in.
x = octane number

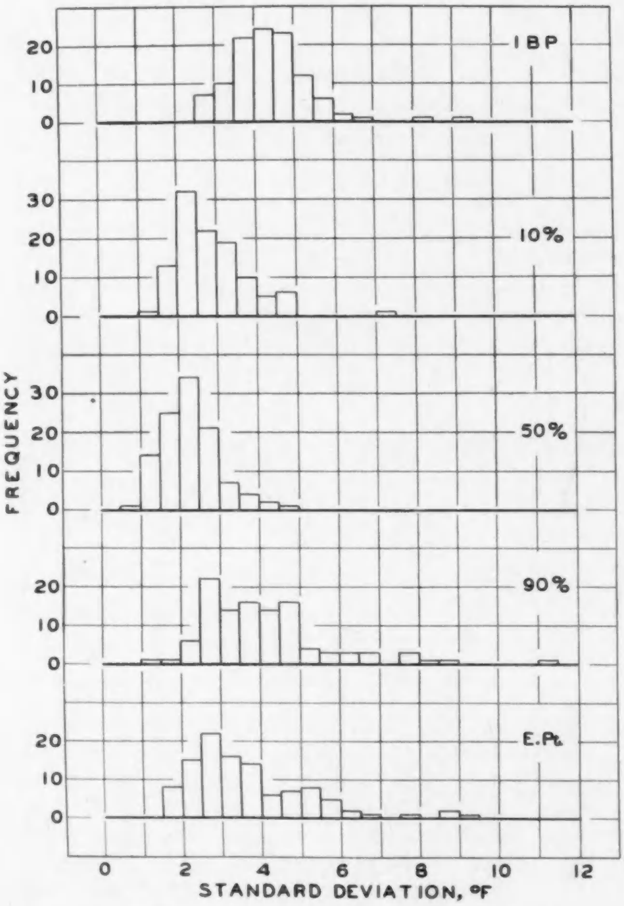
The solid line is the present guide curve for use with 9/16-in. venturi with throttle plate (in use prior to 1942), and is expressed by the equation:

$$y = 0.8839 - 0.0_25330x + 0.0_45075x^2 - 0.0_66667x^3 + 0.0_81250x^4 \quad (2)$$

It is apparent from Fig. 10, especially in view of the tolerance of 0.025 in. allowed in the use of the guide curve, that there is no essential difference in the two curves. In fact, from 40 to 90 octane number they are practically identical, and the largest deviation arises from the fact that the earlier curve, by fitting the data, failed to pass through the reference point at 65 octane number.

In correlating the data used for the 1939 curve, it was necessary to make corrections to the observed compression ratio (micrometer setting) for barometric pressure. This variation, which was also derived from the observed data, was then found to vary with the compression ratio. At a compression ratio of 4.0, the value obtained was identical with that recently found much more accurately by the altitude chamber tests.

Thus the present guide curve and the present correction of the curve for air pressure are substantiated surprisingly well by the data obtained in Exchange Group testing during 1936, 1937, and 1938. The present curve therefore is not in any sense an innovation, but is an exact and continuous representation of the relation found by many years of experience.



■ **Fig. 11 – Distribution of standard deviations of fuel inspection tests**

The distributions of the deviations are normal for the initial boiling point and the 10% and 50% points, but are abnormally broadened for the 90% point and end point, indicating occasional occurrence of unusually large errors

Table 7 – Precision of Inspection Data
Exchange Group – R Samples

Item	Values Covered: Max.	Min.	Mean	Mean Std. Dev.	Max. Std. Dev.
Gravity, deg API	72.9	38.9	60.6	0.24	0.64
Reid Vapor Pressure, psi	13.0	2.4	7.2	0.48*	1.16
Initial Boiling Pt., F	158	82	107	4.5	9.4
10% Recovered, F	210	91	150	3.0	7.4
50% Recovered, F	274	189	231	2.3	4.6
90% Recovered, F	389	233	328	4.4	11.4
End Point, F	452	271	381	3.9	9.0
Recovery, %	98.7	95.2	97.9	0.6	1.1
Loss, %	3.7	0.5	1.2	0.6°	1.3

* Increases with increase in vapor pressure.
° Increases with increase in loss.

XI. Precision of Fuel Inspection Data

Heretofore the triennial analysis has been confined to knock ratings and allied data. The accompanying fuel inspection data have been used only for sample identification. This wealth of data at last proved irresistibly attractive, and it has been included in this analysis.

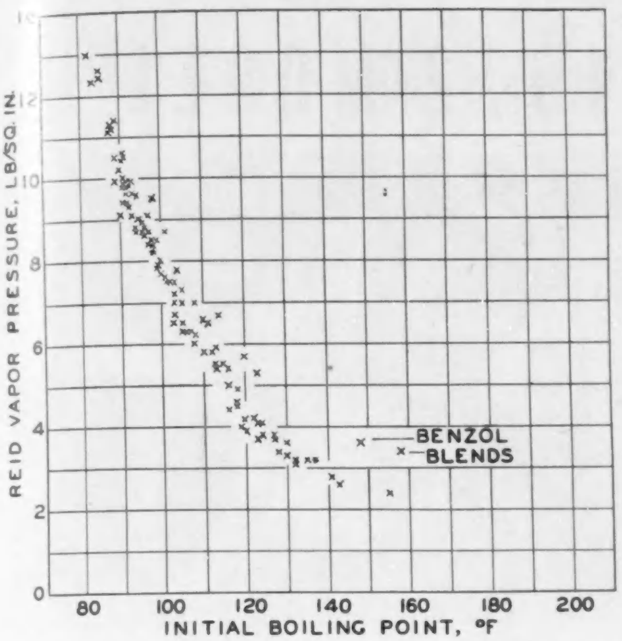
Data on the 109 R-samples only are covered in this part of the investigation. Table 7 summarizes the ranges covered and the precision of each measurement. The mean standard deviation listed in the next to the last column is the root mean square value and, in consequence, is 3 to 10% higher than the arithmetic average of the standard deviations for the individual fuels.

Correlation of the Reid vapor pressure values and their standard deviations showed a definite increase in the standard deviation at higher vapor pressures, the correlation coefficient being +0.330. The probability of this high a coefficient being the result of chance is 1 in 2000. The expected standard deviation consequently varies from 0.32 at zero to 0.51 at 10 psi.

The standard deviation of the loss is closely correlated to the amount of loss, the linear correlation coefficient being +0.803. The calculated average standard deviation varies from 0.40% for a loss of 1% to 1.09% for a loss of 3%. The standard deviation of the recovery is likewise correlated with the magnitude of the recovery, the correlation coefficient being —0.645. A recovery of 99% has a calculated average standard deviation of 0.35%, while one of 96% has 0.95%.

Similar correlations made for each of the temperature measurements showed no relation between the magnitude of the value and the corresponding standard deviation, with the possible exception of the end-point temperature. This correlation disappears, however, if the 8 samples, principally aviation gasolines, with end points below 300 F, are omitted. The remaining samples, having end points from 338 to 452 F, have a mean standard deviation of 3.8 F, with no significant tendency to vary with the end-point temperature.

As shown in Fig. 11, the distributions of the standard



■ Fig. 12 – Correlation of Reid vapor pressure of R-Samples with initial boiling point

deviations for the initial, 10%, and 50% temperatures are normal. Those for the 90% and end-point temperatures, however, show abnormal broadening. In consequence, while the mean values of standard deviations given in Table 7 for the first three of these measurements can be considered true statistics, those for the latter two must be employed with the reservation that, in some cases, values larger than would normally be anticipated may occur. Of opposite effect is the fact that the distillation data have

Table 8 – Precision of Inspection Data on Semi-Annual Samples

Item	Mean Value: Exchange Group	Mean Value: Non-Member	Standard Deviation of One Observation: Exchange Group	Standard Deviation of One Observation: Non-Member
Gravity, deg API	58.86	58.89	0.22	0.23
Reid Vapor Pressure, psi	7.49	7.53	0.44	0.48
Initial Boiling Pt., F	107.2	108.1	4.3	4.8
10% Recovered, F	148.9	149.3	2.6	3.0
50% Recovered, F	229.2	229.3	2.2	2.6
90% Recovered, F	332.5	333.2	3.9	4.1
End Point, F	383.4	383.3	2.9	3.9
Recovery, %	97.77	97.67	0.61	0.62
Loss, %	1.26	1.37	0.60	0.56

not been corrected to standard barometric pressure. As the observed pressures ranged from 28.40 to 30.67 in. hg, some reduction in the standard deviations would be expected to result if correction were made for this factor.

Refinements in the test method have probably resulted in considerable improvement in the precision of determining the initial boiling point. In view of this condition, possible correlations are worth considering. Fig. 12 shows the Reid vapor pressure plotted against the initial boiling point. It is obvious that the two measurements are closely related, although two benzene blends, having initial temperatures of 148 and 158 F, are decidedly off the curve. From a consideration of the respective standard deviations, Table 7, it appears that, in most cases, the Reid vapor pressure could be estimated with fair precision from the initial temperature. The correlation coefficient between initial temperature and Reid vapor pressure is exceptionally high, —0.969, when the two benzene blends are omitted, and the coordinates are transformed to eliminate curvature.

Loss is also correlated closely with the initial boiling point, as shown in Fig. 13. In this case it develops from a consideration of the respective standard deviations that the loss estimated from the initial boiling point has a lower standard deviation than it has from direct measurement.

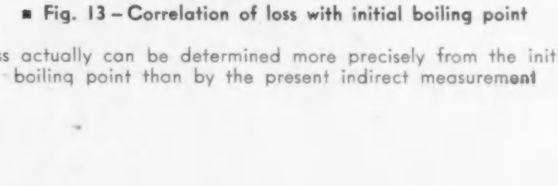
XII. Comparison of Inspection Data Reported by Exchange Group and Non-Member Participants

The composite average inspection data obtained by the Exchange Group and by the non-member participants on the semi-annual samples are given in Table 8, together with the standard deviation for each group.

The two groups agree within experimental error on the values for gravity and vapor pressure. On the distillation data, the Exchange Group values are consistently lower, except for the end point, and this difference is beyond the experimental error except at the 50% point. Also, the Exchange Group value for recovery is definitely higher, and that for loss is definitely lower, than those of the non-member participants. The standard deviations are in general slightly lower for the Exchange Group. All of the foregoing variations can be coordinated by the assumption that, in Exchange Group laboratories, more attention is paid to maintaining the specified temperature in the ASTM condenser and receiver.

XIII. Conclusions

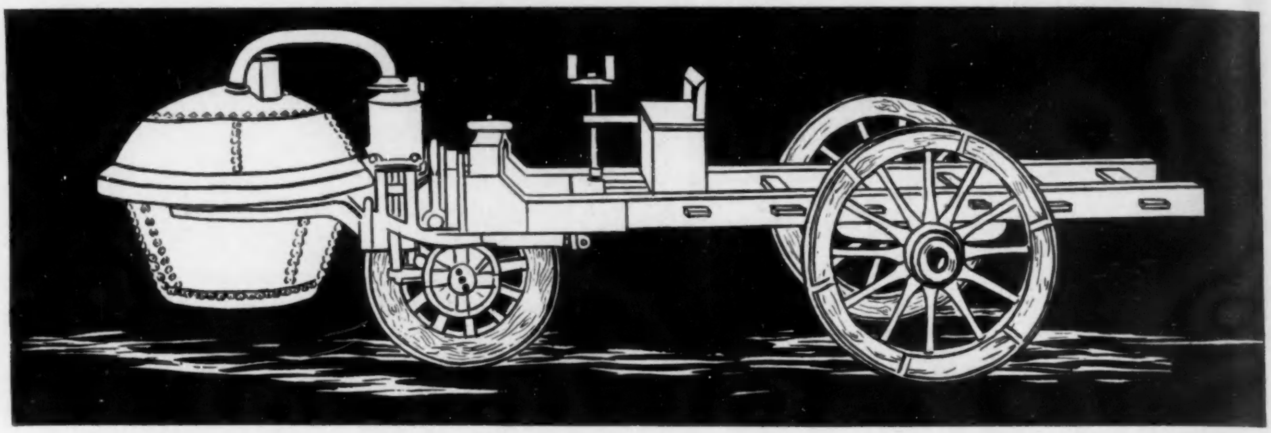
This analysis has shown that the precision of rating by the ASTM Motor Method has continued to improve, and that the precision by the CFR Research Method has greatly improved since its introduction at the end of 1938. The use of the sensitive reference fuel X-1 is probably responsible for at least part of the marked improvement in the precision of ASTM Motor Method ratings of cracked fuels, but the knock rating of X-1 has deteriorated noticeably in the past year. From the performance records of the best laboratories, it appears that the average precision of Exchange Group rating still can be doubled, by careful attention to details. The humidity control apparatus does not influence ratings. The present ASTM guide curve is in excellent agreement with the equation deduced from ratings made in 1936-1938, and derived in the 1939 analysis.



■ Fig. 13 – Correlation of loss with initial boiling point

Loss actually can be determined more precisely from the initial boiling point than by the present indirect measurement

The 1972 MODEL . . .



■ The first motorized military vehicle?

This car, which is thought to have been the first steam carriage ever built, was constructed about 1770 by Joseph Cugnot, a French army captain. Having been built by the authority and with the backing of the war minister, and with military uses in mind, this pioneer vehicle may perhaps be considered as the beginning of the motorized military force

by T. A. BOYD

Research Laboratories Division, General Motors Corp.

AS background for his predictions for the 1972 car, in the early part of this paper, Mr. Boyd looks back one generation to paint a picture of the automotive progress that has culminated in the 1942 models. Probing back through history for examples to back his points, he brings out that virtually all the elements essential to today's automobile were thought of, or came into being one by one, during the long evolution which preceded the actual arrival of the automobile itself. Although the *list* of parts is essentially the same in present cars as in those of thirty years ago, he emphasizes, the parts differ in form, in composition, and in the perfection with which they do their jobs.

Mr. Boyd predicts that the 1972 car will be lighter, and will have a smaller, lighter, more efficient engine located in the rear. Aluminum and magnesium will be used liberally, he says, and rusting will be avoided on ferrous parts. A completely non-shattering, non-splintering glass, much superior to today's "safety" glass is predicted for 1972. Sensational strides in gasoline economy and in fuels themselves are also seen in the next 30 years. Glareless night driving, improved highways, better parking facilities, greater highway safety, and many other kindred advances are seen by Mr. Boyd in his crystal ball.

WHEN all automobile assembly lines were shut down at the end of January, 1942, a total of more than 85,000,000 motor cars had been manufactured in the United States. This present period when no automobiles are being made may be a good time to glance backward over the course of the motor car thus far to see, if possible, something of what its future evolution is likely to be.

In looking backward, it may aid perspective to go into history far enough to trace briefly the genealogy of the automobile, particularly from the viewpoint of seeing just where its essential elements – the differential, the transmission, and so on – came from in the first place. That will take us back a great many years, to be sure. But it will show this interesting and important thing: all, or nearly all, of the essential elements of the automobile were thought of or came into being, one by one, during the long evolution which preceded the arrival of the gasoline automobile itself.

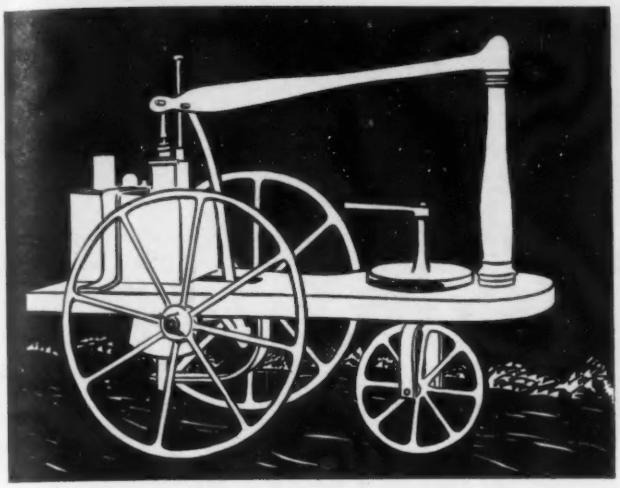
The wish for a carriage to go without horses was a very, very old one. And men of all ages tried to make that wish come true. They built muscle-power carriages, wind-power carriages, spring-power carriages, steam-power carriages, and at last the gasoline-power carriage. During all that time progress was at a snail's pace, of course. But there *was* progress nevertheless. It consisted in the invention, one by one and with long intervals between, of the elemental units out of which the automobile was finally made.

Leonardo da Vinci, for instance, who lived about 450 years ago, made sketches of a spring-power car, and he is

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 12, 1942]

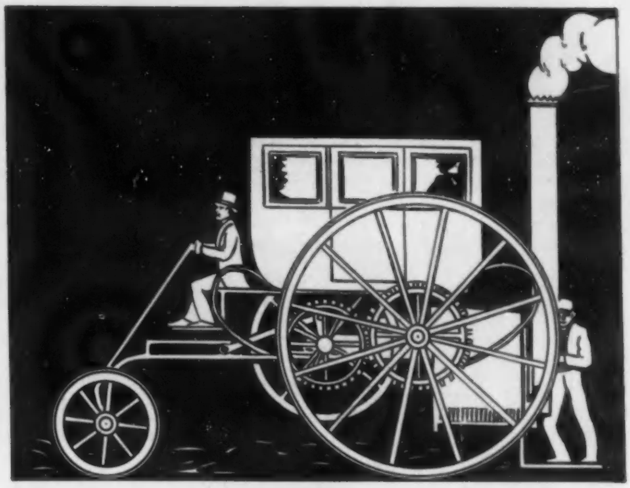
THE AUTHOR: T. A. BOYD (M '27), who has headed the fuel department of General Motors Research Laboratories since 1923, is a co-discoverer of the antiknock effects of the liquid compounds of lead along with C. F. Kettering and Thomas Midgley, Jr. His researches have been concerned chiefly with engine knock. Mr. Boyd is author and co-author of a large number of technical papers, numerous semi-technical or popular articles, and several books. He is an alumnus of Ohio State University, receiving his Bachelor of Chemical Engineering degree in 1918 and the degree of Chemical Engineer in 1938.

I'd Like To Drive It



■ Murdock's steam car, 1784

This little steam-power car was only a model, but an operating one. It was built by William Murdock, associate of the great James Watt of steam-engine fame, and is said to have been the first self-moving vehicle to employ that vital mechanical linkage, the CRANK and the CONNECTING ROD



■ Steam-power carriage built by the inventor of the change-speed gear

In addition to the application of the spur GEAR to the propulsion of a road carriage, as pictured here, Richard Trevithick anticipated the automobile transmission. This he did by conceiving and patenting, 140 years ago, a CHANGE-SPEED GEAR, saying in his patent that "the power of the engine with regard to its convenient application to the carriage may be varied by changing the relative velocity of the road wheels . . . by shifting the gears . . . for others . . . properly adapted to each other"

said to have invented also the *universal joint,* the *bevel* and the *spiral gear,* and the *roller bearing,* all essential elements of the automobile. The first *internal-combustion engine,* believed to have been built in 1680 by Christian Huyghens and to have used gunpowder as fuel, made what is thought to have been the first use of the *cylinder* and *piston.* The *pneumatic tire* was invented and patented in 1845 by the Englishman, Robert William Thompson. In 1862, the Frenchman, Beau de Rochas, conceived and patented the *four-stroke cycle* and, in 1876, the German, N. A. Otto, built a *gasoline engine* utilizing that cycle, which has since been called the Otto cycle. The *gasoline* and *oil* to run the gasoline engine on had become available after 1859, when Edwin L. Drake drilled the first oil well at Titusville, Pa.

When to the several elements just enumerated are added all those mentioned in the legends of the accompanying illustrations, the list of elements essential to the motor car invented or used before the automobile pioneers of the 1880's and 1890's began their work becomes somewhat as follows:

- Gasoline engine, with
  - Cylinder and piston
  - Crankshaft and connecting rod
  - Valves, carburetor
  - Electric ignition, spark plug, distributor
  - Four-stroke cycle, and compression for power
- Wheel, and pneumatic tire
- Steering wheel, and stub-axle steering
- Gear, and sliding-gear transmission
- Differential
- Leaf and spiral springs
- Gasoline and lubricating oil

It is seen that this list of the elements thought of or used before the automobile, as we know it, came into being, includes almost all those needed out of which to make an automobile.

■ What Our Automobile Pioneers Did

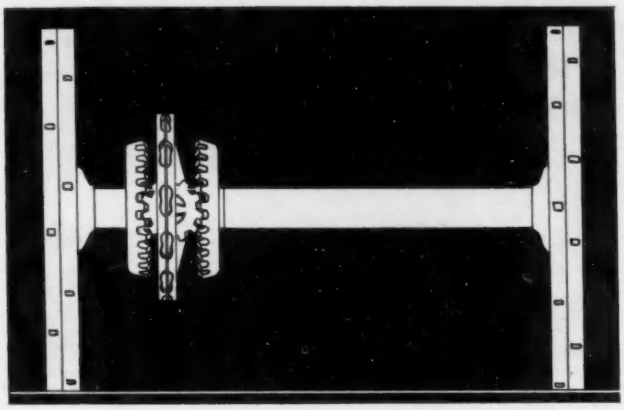
All this does not, however, detract in any wise from the greatness of what was done by our automobile pioneers – by those men who built the first successful cars. They surely had plenty to do in putting those embryonic elements together into a car that would work. First of all, they had to have faith enough in the horseless carriage as a means of transportation to persist in building them in spite of having been called crazy again and again. To do that they had to get the needed elements. And although, as just shown, most of those elements had been invented or used before, it was not possible to buy many of them from suppliers. Some even had to be invented all over again, the original invention having been long forgotten. However, the contribution of the automobile pioneers was primarily in engineering, not in invention.

It took several years to learn how to make cars that were good enough to live down the wisecrack of the time about the car with wooden wheels, wooden axles, wooden body, wouldn't run. So often those early cars would not run that their most useful accessory was a tow rope. But, by the year 1911, cars that ran pretty well were being made. One of their principal deficiencies was that they

had to be started by hand cranking, which was not only troublesome, but sometimes dangerous also. This deficiency was first overcome in a practical way by the self-starter which appeared on the 1912 Cadillac car, just 30 years ago.

■ Advances in the Generation Just Past

It has been pointed out that, if a 1912 model car and a 1942 car were each torn down and the parts laid out side by side, the two would be seen to have pretty nearly the same parts. Both have pistons, connecting rods, differentials, tires, and so on. But although the parts *list* is much the same in the two instances, the parts themselves are not really alike. They differ in size, in form, in composition, and in the perfection with which they do their job. For the 1942 car has been marvelously improved over the 1912 model.



■ **Pecqueur's differential**

Onésime Pecqueur took out a patent in 1828 in which he illustrated and described the DIFFERENTIAL, pictured here, for permitting each driving wheel to turn independently. As illustrated, Pecqueur's differential was arranged for the power to be received through a sprocket chain, but he mentioned that it could also be applied through gears, as is done today

In this improvement there have been several main trends. Some of these are the trend toward more universal usability, greater dependability, better roadability, greater comfort, and better appearance; the trend toward lower cost; the trend toward higher power and better performance; the trend toward more ton-miles per gallon.

The trend toward more universal usability, for instance, to which the self-starter made such an important contribution 30 years ago, continued upward as other advances were made from time to time. It used to be customary to lay up the car in the winter time, of course. But, thanks to the low-cost closed body, to the paved road, to gasolines and lubricating oils favorable to ready starting in cold weather, and to a host of other improvements, cars are driven in winter now just the same as in summer.

In respect to the trend toward lower cost, which is a particularly important one, a car of given wheelbase and weight costs less than half as much when sales were discontinued last year as did a car of the same size in 1920. In dollars per horsepower, cars cost only about one-third as much then as they did in 1925. And, furthermore, it costs the owner not much more than half as much to run his car now as it did 15 years ago.

When next you hear the statement – made so often in recent years – that machinery and improved methods of manufacture, which have done so much to make possible those reductions in cost, threw men out of work in the automobile industry, do not believe it. It is only necessary to look up the figures of the U. S. Department of Labor to see that the reverse is true. In 1940, for instance, the most recent of what might be considered an approximately normal year, automobile factories employed the same number of workers as in 1929. But in that year of 1940 only 83% as many cars were made as in 1929. Because the increasing size and complexity of cars more than offset the effects of improvements in manufacturing methods, more men, not fewer, were employed per car in 1940 than in 1929, and they made higher wages too.

In the trend toward higher power and better performance, the power of an engine of given size or cubic displacement has been almost doubled during the past 20 years. When, about 1924, the General Motors Proving Ground was established, there was provided there a test hill of 11.6% uniform grade. The test of the hill-climbing ability of a car was to start up that hill from an initial speed of 10 mph, and see how far up the course it could go in high gear. But long ago that hill lost much of its value, because all cars could go clear to the top in high. And so the criterion of hill-climbing ability was changed to the *speed* at which the car went over the top from a 10-mph start. In that test now, cars accelerate right up the hill and go over the top at a high speed. This gain in performance or in engine power is, incidentally, one of the things that has made mechanized military equipment so practical.

In this respect, as well as in others, the improvement of the automobile has been made to follow the course of public acceptance. It was thus because of public demand that automobile engineers have had to make the goal of more miles per gallon secondary to what is called higher performance – more zip at the traffic light. The flash of green in the semaphore has started more races already than the crack of the pistol ever will. Nevertheless, in spite of this handicap, ton-miles per gallon have, within the past ten years, been improved by 20% or more. But now, fortunately, there are beginning to be signs that, in the future, people may place more emphasis upon economy than upon still higher performance.

■ What's Ahead?

Turning now, and, with the aid of the perspective gained by looking backward, peering up the road toward the future, what do we see? I am asking this question much as did Sam Goldwyn, who at a Hollywood producers' meeting jumped up and said: "Gentlemen, for your information I would like to ask a question."

Since the trends in developments during the 30 years from the time of the self-starter have just been traced, we might try peering ahead 30 years also to 1972. No one has enough imagination to foresee just what will be accomplished in the generation just ahead, of course. But it might help imagination a little to suppose that, through some magic, time has all at once run forward 30 years

to the spring of 1972, and that, from the vantage point of that time, we are looking backward to 1942.

The first thing recalled as we look back is that there really was no automobile industry then. The great assembly lines, many of them, had been torn out and the space was given over to making engines of destruction. It was fortunate, though, that the special skills and knowledge of the highly specialized method of manufacture called mass production, as developed within an industry which had already manufactured more than 85,000,000 motor cars, made it possible to prepare quickly for the job of fighting the powerful aggressors raging in the world then.

Those troublous times finally came to an end, though, as all things must. And, when the war did end, much of the knowledge gained during its progress was found useful, just as the developments in nitrocellulose and in solvents for it, made during the World War of the previous generation, helped to make practical the synthetic finishes which began to be used in the early 1920's. The large participation of the automobile industry in the aircraft field during the war in progress in 1942 was one thing in particular that produced big changes in the automobile.

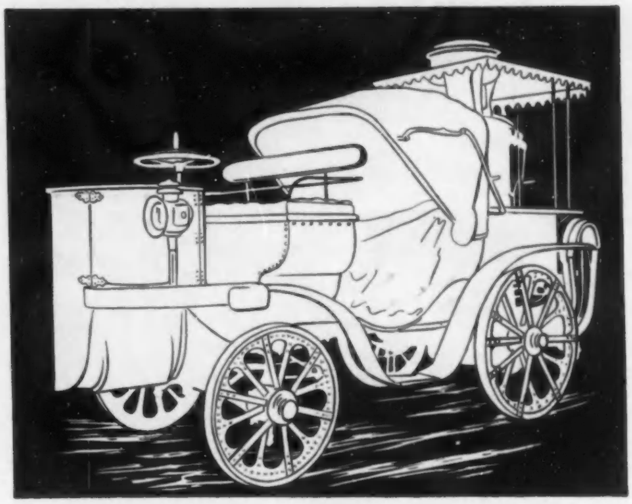
As we look back at the 1942 model car with our 1972 eyes, we are amazed at how heavy it was. Cars then weighed 3500 to 4500 lb – and that 4000 lb of metal, rubber, and glass were often occupied by only one person. One of the main reasons why the cars back in 1942 were so heavy was that, to give the kind of performance desired, the engine had to be such a big one. Some of those 1942 engines weighed as much as 800 lb or more, or about 6 lb per hp of output. So engines were always located in the front then. There was much talk about putting engines in the rear, to be sure. But, as a matter of fact, the engine had to be in front because, for one thing, it was so heavy that, if put in the rear, the car could not be handled properly at high speeds. Also, so much of the heat of the fuel had to be taken out through the radiator then that, unless the radiator was up in front to get the benefit of natural draft, the power required to drive the cooling fan was prohibitive. Believe it or not, as much as a third of the heat of the fuel was wasted through the cooling system back in those days.

Very little aluminum or other light metal was used in cars then, for as yet no one had found how to produce aluminum and magnesium cheaply enough. Aluminum cost 16 to 18c per lb then, and magnesium was still more costly. So bodies and fenders were all made of sheet steel, about 1/30th of an inch thick, supported on steel frames. Contrary to conditions at present – that is, in 1972 – rusting of fenders and bodies was a serious problem then, particularly in regions where salt was spread on the streets and the roads to melt snow or to lay the dust. Sometimes fenders would actually rust clear through. How different all that has been ever since a really effective means was found for keeping oxygen from devouring iron.

The glass in the windows of cars then was called "safety" glass; but, by comparison with the altogether non-shattering, non-splintering glass of 1972, it was really not very safe. The glass they had back there was a great improvement over plain glass, to be sure. But calling it "safety" glass was really not much more fitting than it was for Otto to name the gasoline engine that he built away back in 1876, the "silent" engine.

The cars people drove in 1942 gave less than 20 mpg of gasoline – usually only 12 to 18 mpg. At that rate, strange as it seems to us in 1972, only 10% of the heat of the gasoline was being converted into push for the car. So, in spite of the low cost of gasoline per gallon then – it cost less than 20c, and 6c of that was tax – the owner of a car during its lifetime or period of service paid out as much for gasoline and oil as the car itself cost in the first place. Thus, although a great deal of progress had been made, even then, toward reducing the cost of cars and the cost of operating them, a government survey made about that time showed that from 7 to 12% of family income was being spent on automobile transportation. That was a bigger chunk of the family income than was spent for clothes for the family, and it was two-thirds as big as that expended upon housing[1]. Thus it appears that, among the many great advances made in cars since 1942, the biggest of all has been that of lowering the cost of owning and driving a car.

The gasoline on which cars were being run in 1942 was somewhat below 75 octane number, and ten years earlier still octane number had not risen above 60. Already, then, to be sure, they knew something about the value of higher octane gasolines, such as that we have now in 1972; and gasolines of fairly high octane numbers – 100 and even more – were being made in limited amounts for use in aviation engines. In fact, the war in progress then was demanding for airplane engines huge amounts of 100-octane gasoline. But in 1942 it cost twice as much to make 100-octane gasoline as to produce the 75-octane gasoline used in cars. And so, with most automobile engines then using gasoline of 75 octane number or less, the maxi-



■ Steam carriage which foreshadowed the automobile

The steam road carriages of the Frenchman, Amadée Bollée, built in the 1870's, had lines and construction similar to the present-day automobile. They had ENGINE IN FRONT, HORIZONTAL DRIVE-SHAFT, DIFFERENTIAL GEARING, and front wheels swiveled on STUB AXLES for steering. Note also the STEERING WHEEL and the HEADLIGHTS

[1] See Miscellaneous Publication No. 415, U. S. Department of Agriculture in cooperation with the WPA, 1941: "Family Expenditures for Automobile and Other Transportation."

mum output in brake mean effective pressure was only 110 to 120 psi, which was only one-half the output of the airplane engines of that time, and very much below that of our 1972 model automobile engines. Back in 1942 there was also an engine problem called roughness which was little understood as yet and which increased so fast in seriousness with rising pressure that some engineers thought cylinder pressures such as we are using in our highly efficient 1972 model engines would never be practical.

Not at all understood either, back there in 1942, was the important subject of just how gasoline burns in the engine to produce power. And that condition was true in spite of the fact that practically all automotive transportation depended upon that event, and that a great deal of research on it had already been done. The good understanding of engine combustion that we have now, in 1972, is due to the determination and persistence of those who realized that the fundamental truth about engine combustion was an item of knowledge essential to the most economical utilization of the combustion process.

Another reason why in 1942 cars ran only 12 to 18 mpg was that most of the time engines had to be operated then at only a fraction of their capacities. Thus, at a level-road car speed of 30 mph, that fraction might amount merely to one-seventh of the full engine power at the rpm corresponding to that car speed. Under such low-throttle conditions, as much energy was consumed in engine friction and pumping loss as was delivered to the car as power. This condition was brought about in large part by limitations imposed by the transmissions and the axles of the time, which fixed engine speed so inflexibly to the road speed of the car that mostly the engine was running under conditions of speed and load far removed from its most economical range.

Another one of the problems they had back there in 1942, which we are glad has been solved since, was how to get about at night without glaring other travelers off the road. In those days, they relied almost altogether upon headlights carried on the car itself. But the trouble was that they could not make those headlights bright enough to see the road properly without causing discomfort and annoyance to the drivers of other cars on the road. The lights they did have on cars then were not as effective as they might have been either, because the character of the road surface and the color of it – many roads were black – did nothing toward intensifying visibility with the light falling on it.

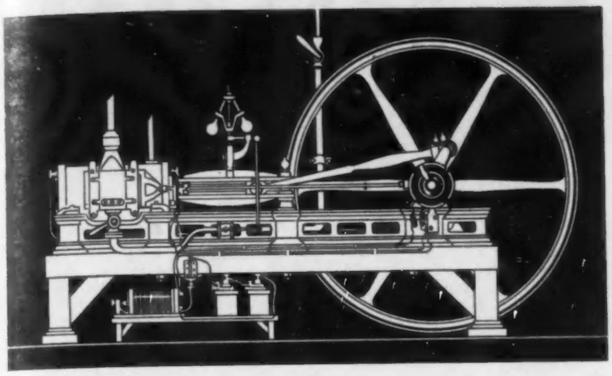
■ Sadi Carnot and his classical work

In his little book, "Reflections on the Motive Power of Heat," printed in 1824, Sadi Carnot suggested the HIGH-COMPRESSION, INTERNAL-COMBUSTION ENGINE. Although his treatise was devoted to the steam engine for the most part, Carnot wrote this there also: "Air can be heated directly by combustion within itself. Thus a considerable loss is avoided . . . In order to produce a great expansion of the air, . . . it would be necessary to subject it in the first place to a . . . high pressure"

■ Here Goodyear discovered how to vulcanize rubber

The rubber tire, so essential to the automobile, is possible only because, after long search, Charles Goodyear found that mixing rubber with sulfur and then heating the mixture changes "gum elastic" from a useless stuff, sticky and runny in summer and hard and brittle in winter, into a marvelously resilient and durable substance. Goodyear's discovery, made in 1839, allowed Robert William Thompson to invent the PNEUMATIC TIRE in 1845, an idea more successfully revived in 1888 by John Boyd Dunlop. This is a present-day picture of the historic house in Woburn, Mass., where, in the room that was then the kitchen, Goodyear made his great discovery 103 years ago

Then, too, the roads and city streets which they had in 1942 were many of them still of the "horse-and-buggy" variety. Some roads were still crooked and narrow. They had as yet found no means of building good secondary roads at low cost. Even on heavily traveled highways, there were very few underpasses or overpasses. At railroads, too, there were so many grade crossings that over 5000 people were killed each year trying to cross them. Limited-way roads on which cars could run at a good speed without interruption were almost non-existent at that time. At intersections of traveled highways and streets there was almost always a system of time-operated traffic lights which allowed traffic to flow in one direction for a few seconds only. Then the stream had to be stopped to let the traffic crossing the other way flow for an equal number of seconds. So there was much starting and stopping of cars, which was very hard on brakes and on miles per gallon, and in which immense amounts of time was wasted standing at traffic lights. This was particularly



■ **A gas engine of the 1860's**

Building in part upon prior progress, the Frenchman, Lenoir, began in the early 1860's to manufacture internal-combustion engines, and sold three or four hundred of them. The double-acting Lenoir engine pictured here had ELECTRIC IGNITION, two SPARK PLUGS, and a DISTRIBUTOR. Note also the FLYWHEEL. However, Lenoir's engine did not compress the charge before igniting it, as 40 years before Carnot had said was necessary for power and efficiency

true during the morning and the early evening traffic hours.

Where to park cars in the congested areas of that time was one of the biggest problems drivers had to contend with. And that was so in spite of the fact that, on many of the streets covered with high-cost paving, a line of cars was parked along each curb, taking up half of the highway sometimes, and so making a huge investment only 50% useful.

There was then also the problem of what was incorrectly called "safety" on the streets, but which at that time should in reality have been called "danger." In 1941, more than 275 people were killed in automobile accidents on the streets of Detroit alone, and several times that many were injured. The automobile itself was not responsible for much of this, to be sure, and only a third of those killed were riding in the cars involved. Most were pedestrians. One of the changes made since 1942 which has done much to banish peril from the streets was to set up the effective means that we have today of teaching people how to drive cars and of keeping those who cannot drive safely from getting behind the steering wheel. Back there in 1942 too, roads and city streets were so poorly planned, or so badly adapted to safe and speedy automobile traffic, that some of the biggest advances of all during the past 30 years have been those made in highway and traffic engineering.

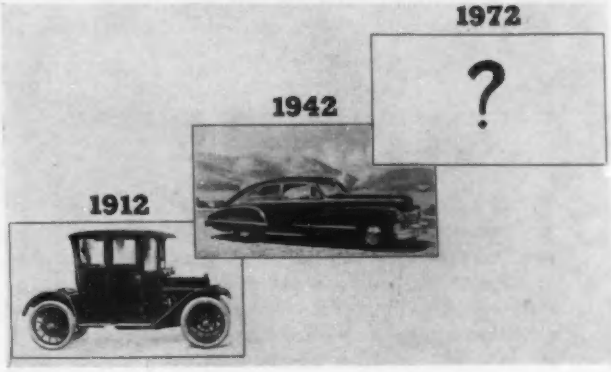
■ Back Again to 1942

At this point we must come back from our imaginary transmigration to the realm of 1972 to the present. What has just been related has not been said with the expectation of giving in any wise a complete catalog of the improvements in cars waiting to be made, of course. Nor has any of it been said with the intention of criticizing the automobile of the present, for it is a marvelous thing. The aim has been only to point out, as concretely as possible, just a few of the many respects in which the automobile might still be improved, *if only we knew how*.

We have now a limited amount of information. Upon that knowledge has been built a technique of engineering and a line of automotive products which are extremely useful and very successful. If we had more knowledge, it ought to be possible to build still better automotive products. And surely we are going to get more knowledge – a great deal more knowledge – for, in reality, very little is known as yet.

If poetry is not out of place in such a paper as this, four lines from a poem of Alfred Lord Tennyson's may here be quoted. Tennyson called this poem *Mechanophilus,* and in it he described the condition of knowledge in his day, which condition is believed to be true still, particularly as relating to the motor car:

> As we surpass our fathers' skill,
> Our sons will shame our own;
> A thousand things are hidden still,
> And not one hundred known.



■ **Upward from 1912**

In the left box is the car on which the self-starter appeared first. The product evolved through the 30 years of development since the arrival of the self-starter is pictured in the middle box. But the 1972 model toward which this paper projects can only be represented by a question mark

Our groping for new knowledge is in slow motion, to be sure. But, once a new fact has been laid hold of and pulled up out of the murk of ignorance, it is never forgotten. The process of discovery has a ratchet on it – it is non-reversible. Thus those who carry on into the future – and that includes all *young* engineers and men of science – will have all the old knowledge as well as the new to work with.

So, then, the answer to the question of what's ahead for the motor car is a confident expectation of much further progress. That prospect is so alluring as to make me wish that I, for one, could still be around 30 years hence to drive the 1972 model. It ought to be a better car, a much more efficient one, and a still less costly one.

■ Acknowledgment

Grateful acknowledgment is made of the help of E. W. Scanes in preparing illustrations for this paper.

PRODUCTION BREAKDOWN

PRODUCTION breakdown illustration is a device or method for speeding the production of airplanes. Breakdown illustrations depict in simple, easily understood drawings the operations necessary to assemble each component part of an airplane structure. The average mechanic, after a few moments' study of this drawing, understands precisely what the results of his work will look like and how to achieve those results.

The rapidly expanding aircraft industry has the vital and immediate problem of training the greatest number of men in the shortest possible time. We are training many thousands of mechanics and artisans with the aid of the public-school system and in company training centers. However, it is impossible to teach all of these workmen, coming from all walks of life, to be mechanics, or even to read blueprints, in the time available. Consequently, the Douglas Aircraft Co. has devised a means of entirely eliminating the use of blueprints on the final assembly lines.

A group of specialized engineer-artists, known as production illustrators, has been created for the purpose of illustrating the work to be done by each man in every position on the line. These illustrations are known as "Illustration Job Tickets," and give each man a clear picture of the local structure as he will see it on the airplane, showing where the sub-assemblies and parts are to be at-

THE primary purpose of "Production Breakdown Illustration" is to speed the production of military aircraft by giving each workman a simple picture of the part or assembly upon which he is working, together with an easily understood description of the operations and tools necessary to do his job. It is a modern adaptation of the ancient device of transmitting thought by pictorial representations, used by man through the centuries.

As developed at the Douglas Aircraft Co., the use of production illustrations has proved of tremendous value not only to the mechanic on the production line but in the perfecting of designs for new type airplanes, the more efficient use of the production line technique, the more accurate determination of material needs, the speedier development of tooling and jigging requirements, and in other phases of the manufacturing process.

Production illustration routines now fall into four major phases. First, during the design stage when innumerable new problems arise as the result of the demand for better airplane performance, the illustrations facilitate closer coordination among the various specialists who have the responsibility of solving these problems. Second, after the design is determined, the structural units making up the airplane are broken down into drawings of manufacturing sub-assemblies and of the various functional installation such as "flight controls," "hydraulic system," and so on. Third, the fabricating and assembly operations are systematized and depicted on illustrated "job tickets" which give all information necessary to do the job illustrated by the ticket. "Sub-assembly sketches" are made to illustrate the bench fabrication of assemblies or sub-assemblies and, with the job tickets, to guide the routing of sub-assemblies, material, and parts. Fourth, changes and adjustments are made to correct remaining deficiencies in production routines.

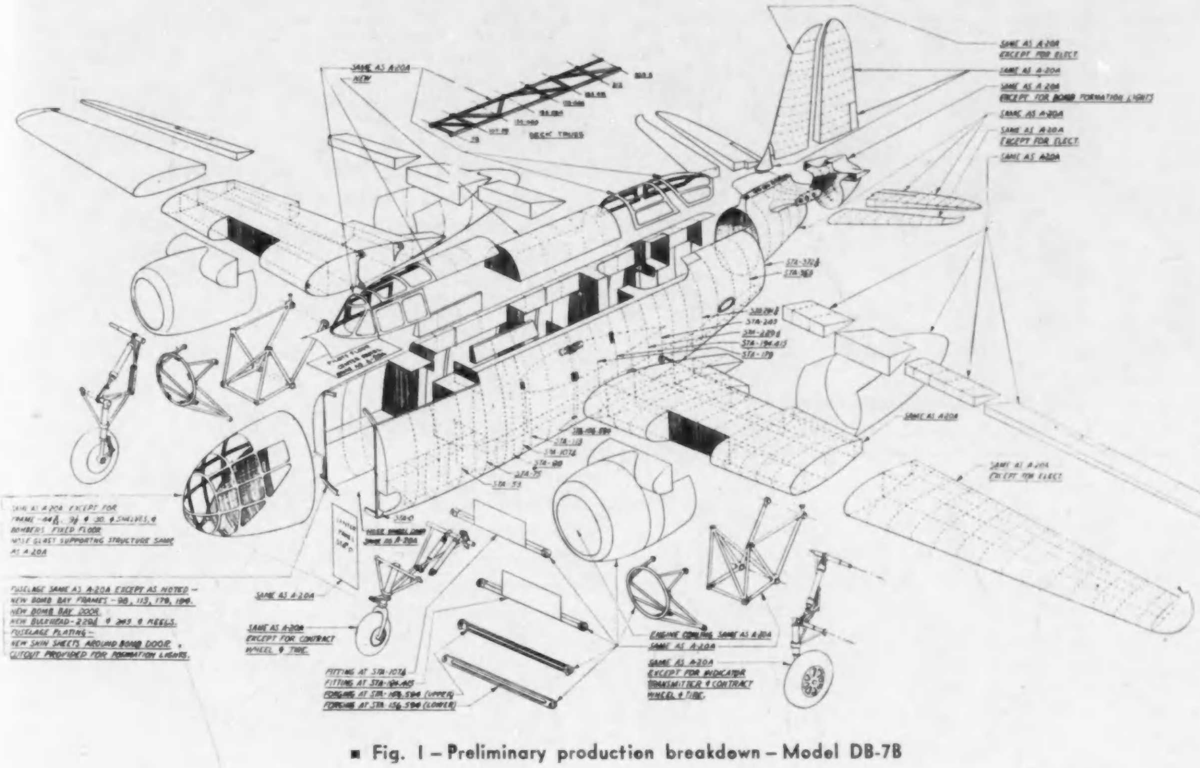
Upon completion of these four phases, the entire manufacturing process is frozen and, except for minor adjustments, production follows the routine procedures determined upon and fully illustrated in the production illustrations.

★ ★ ★

THE AUTHOR: GEORGE THARRATT (M'42), chief engineer of the Adel Precision Products Corp., was born in Hull, England, in 1894, of Scottish parents. He attended elementary schools there; then four years at Hull Technical College followed. Coming to this country in 1930 as study engineer with Glenn L. Martin Co., he did much of the preliminary design work on the first China Clippers. Later he was associated with Great Lakes Co. as project engineer on their dive bomber; with North American in California; and with Douglas Aircraft – where he remained in the preliminary design room until 1939 when he organized the new Production and Illustration Department, acting as its coordinating supervisor. After being wounded in France during World War I, he was transferred to service training of new mechanics for the Royal Air Force. The Air Ministry at that time employed profusely illustrated handbooks for instructors in this work. The value of illustrations in training mechanics was so indelibly imprinted on Mr. Tharratt's mind in this work that out of this experience was gradually evolved the "Production Breakdown Illustration" development.

ILLUSTRATION by GEORGE THARRATT

Chief Engineer, Adel Precision Products Corp.,
Formerly Coordinating Supervisor,
Production Illustration, Douglas Aircraft Co., Inc.



■ Fig. 1 – Preliminary production breakdown – Model DB-7B

tached in accordance with a numerical operation sequence. A list of the necessary bolts, nuts, screws, clips, and so on, to attach these parts, and a list of the tools necessary to do the job, are included on the drawing.

A complete series of "Illustrated Job Tickets" is based on an elapsed time schedule for the complete assembly line. Competent development men, working in conjunction with the tooling department and the production illustrators, establish a definite working time element for the assembly line as a whole. The value of establishing this time standard or base is at once apparent.

Of course, blueprints cannot be eliminated in any department where precision parts are to be manufactured, nor can these illustrations ever take the place of the engineering drawings. The "Production Illustration Group" reads and analyzes the blueprints for the purpose of explaining by means of perspective sketches these engineering drawings for the benefit of the men who cannot read them for themselves. Thus, the possibility of mistakes is eliminated, thereby saving the time of valuable supervisors and leadmen who otherwise would find it necessary to explain the job to each workman many times over. It also affords some guarantee that the job will be done in the best and quickest manner, since the procedures have been originated by experienced mechanics, each of whom is a specialist in his line.

The earliest known uses of the sketch for explanatory purposes were the drawings of the early cave dwellers. They, having no written language, were forced to use the only method possible to record their thoughts and actions, thus enabling us to get a valuable insight into their life and activities which, otherwise, would have been unknown to posterity.

The Egyptians and the American Indians used the pictorial, descriptive method of recording their history. The early Chinese, with their usual perspicacity, understood the power of the sketch. Confucius hit the nail on the head when he said: "A picture is worth 10,000 words."

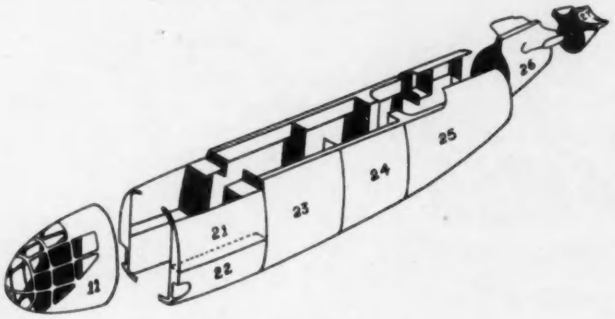
Leonardo da Vinci, the famous Italian scientist, inventor, artist, and sculptor, knew the value of the perspective sketch. He used it almost exclusively for depicting and explaining his ideas and inventions. Born in 1452, he was already famous when Columbus discovered America. An excerpt from the Encyclopaedia Britannica says: "History tells of no man gifted in the same degree as Leonardo was at once for art and science. In art, he was an inheritor and perfecter, born in a day of great and many sided endeavours on which he put the crown, surpassing both

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Oct. 31, 1941.]

predecessors and contemporaries. In science, on the other hand, he was a pioneer, working wholly for the future, and in great part alone. That the two stupendous gifts should in some degree neutralize each other was inevitable."

Thus, it is at once apparent that the pictorial definition of mechanical problems is as old as history.

The outstanding profession where artistic rendering has



■ Fig. 2 – Zone chart – fuselage

been used consistently is architecture. It has always been necessary to have a picture of the finished structure. For example, in city planning the blending of architectural styles with the existing surroundings is necessary. Many other reasons too numerous to mention here require that a completed conception of the entire plan be reduced to black and white, or to color, on the drawing board.

World War I proved the value of the engineer-artist. In speaking of an engineer-artist, reference is made in terms of relative values, that is, the engineering knowledge should be predominant, with the artistic skill as a secondary function. From August, 1914, to the middle of 1916, the war was fought in the open. Then it bogged down to trench warfare. But on Sept. 15, 1916, *mechanical warfare* was introduced by the British tanks at the insistence of Winston Churchill, and contrary to the wishes of the military experts. Proving an instant success, they were put on a production basis immediately and a training program for skilled operating personnel was inaugurated.

Parallel to the development of tank warfare arose the necessity for training a vast army of mechanics for the rapidly expanding flying services, "The Royal Naval Air Service" and "The Royal Flying Corps," later combined and known as the "Royal Air Force." Typical of the lack of interest with which aviation was viewed was the author's experience. Wounded while serving with the artillery in France and considered to be of no further use for "active" service, he was promptly transferred to the aviation forces. It is known that, for every airplane in the air, at least ten mechanics are needed on the ground and to find enough men with sufficient mechanical knowledge to take care of the ground work, was a very serious problem, especially when every available skilled mechanic was needed in the factories. Skilled instructors also were scarce, for the same reason. Consequently, some means had to be devised to overcome this serious shortage. The author was transferred to a pilot training station in England where a large repair shop was being organized and, on arrival, found that very few real mechanics were left due to the heavy demands overseas.

For the most part these men left behind were wounded transfers from the infantry and artillery, and had very meager mechanical knowledge. The Air Ministry apparently had anticipated this need for training and was supplying profusely illustrated handbooks to enable instructors to absorb the necessary knowledge quickly and in a thorough manner. It was possible, through the medium of these reference charts, diagrams, and illustrations, to instruct the new mechanics in the use of various tools and machinery for the repair and maintenance of all types of airplanes. Each engine had a completely illustrated book of instructions which even the poorest mechanic could understand readily. Every type of airplane, too, had a book of instructions, complete with illustrations showing every phase of assembly and rigging. And some of those early "bird-cage" designs needed very expert rigging indeed.

The value of illustrations was so indelibly imprinted on the author's mind, that they were used exclusively for classroom and repair-shop instruction for the station mechanics.

The development of production illustration work at the Douglas plant may prove of interest. The preliminary design department of the Douglas Aircraft Co., received great encouragement in pictorial representation for Army and Navy bids from A. E. Raymond and E. F. Burton, who were at that time chief engineer and chief designer, respectively. In August, 1939, a complete picturization of the proposed manufacturing breakdown and its progressive, detailed fabrication, plus a position breakdown plan, on a mechanically moving assembly line, was included for the first time in a military airplane bid. It received very favorable comment from Washington and proved a boon to the Production Design and Tooling Departments.

In anticipation of the present rush of war orders, with the resultant expansion, mechanically moving lines, and lack of trained personnel, it was agreed that a group be immediately organized to be known as the "Production Illustration Department." The first duty assigned to that department was to study and break down into manufacturing production units the DB-7 type twin-engined attack bombers. The results of our work were to be used by engineering, tooling, and shop personnel. At that time the fuselage was being fabricated as a complete unit and difficulty was being experienced with installations on the final assembly line due to the very restricted working space in such a small fuselage.

Perspective sketches were made showing suggested remedies for this condition and, after several conferences with the engineering and tool design groups, it was decided that the fuselage be split vertically on the centerline and fabricated as right and left halves. See Fig. 1, preliminary production breakdown diagram. This arrangement would allow more men to work on fuselage fabrication with greater freedom of movement. The same advantage would also apply to the assembly line where the two halves received their respective installations. It developed that nearly 80% of the internal equipment and functional installations could be installed before the two halves of the fuselage reached their joining positions on the assembly line.

It was now necessary for the manufacturing division to

estimate how many installation positions would be required on the final assembly line due to this change, in order to produce the required output of parts and assemblies. A development group was organized consisting of the supervisor and his assistant for that model, plus several leadmen who were specialists in their particular line. It was their responsibility to break down the information given on engineering blueprints to determine the installations for each assembly-line position, to approximate the amount of work to be done in each position on an elapsed time basis, and to make suggestions for the necessary drill jigs, templates, special tools, and so on.

From their findings an organized line was established, and the necessary floor space, storage facilities, and handling fixtures were determined. At the same time, the production illustration group was busy coordinating this information by means of sketches, thereby promoting more rapid decisions through correct and simultaneous visualization of the general problems.

The fuselage halves on the assembly line were divided into *working zones* for the purpose of illustrating in detail the work to be done in these zones (Fig. 2) at every position on the assembly line and, for this purpose, the development men supplied all the necessary information for inclusion on each zone-position illustration, as follows:

(A) All part numbers and a description of their installation.

(B) All component attaching parts, such as bolts, nuts, washers, cotters, and so on.

(C) All jigs, templates, and special tools used.

(D) Approximate estimated time based on previous experience.

(E) A list of company and personal tools required to do the job.

This information was coordinated and checked with the planning department to provide for the routing of parts and sub-assemblies to their proper position on the assembly line.

These trial production illustrations proved to be of such inestimable value as an organizing and visual procedure function that the system was incorporated into the basic layout of the new Douglas Long Beach Plant. From this experimental beginning a definite procedure has been established for all models in all manufacturing divisions which is best illustrated in the graphical form shown in Fig. 3.

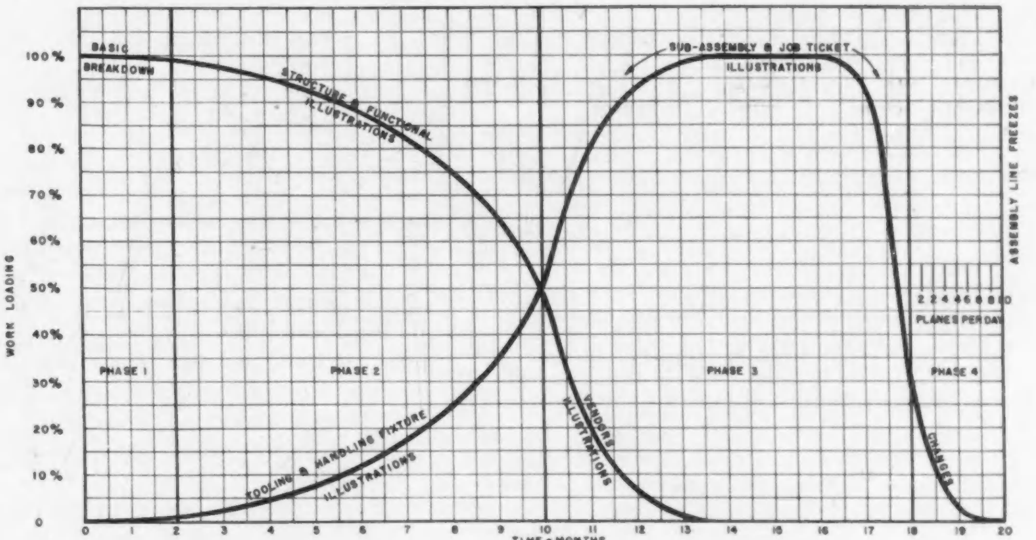
The *work-loading graph* between the preliminary design stage and actual freezing of production techniques on the final assembly line has two distinct curves. One is for *engineering* which starts at 100% and reaches zero at the middle of shop development. The other is for *manufacturing* which starts at zero and reaches 100% at the same point, returning to zero at the point of freezing on the final assembly line.

Today the work of the Production Illustration Breakdown Department is divided into four distinct but overlapping phases, each of which is the product of past experience and each of which must be considered in turn:

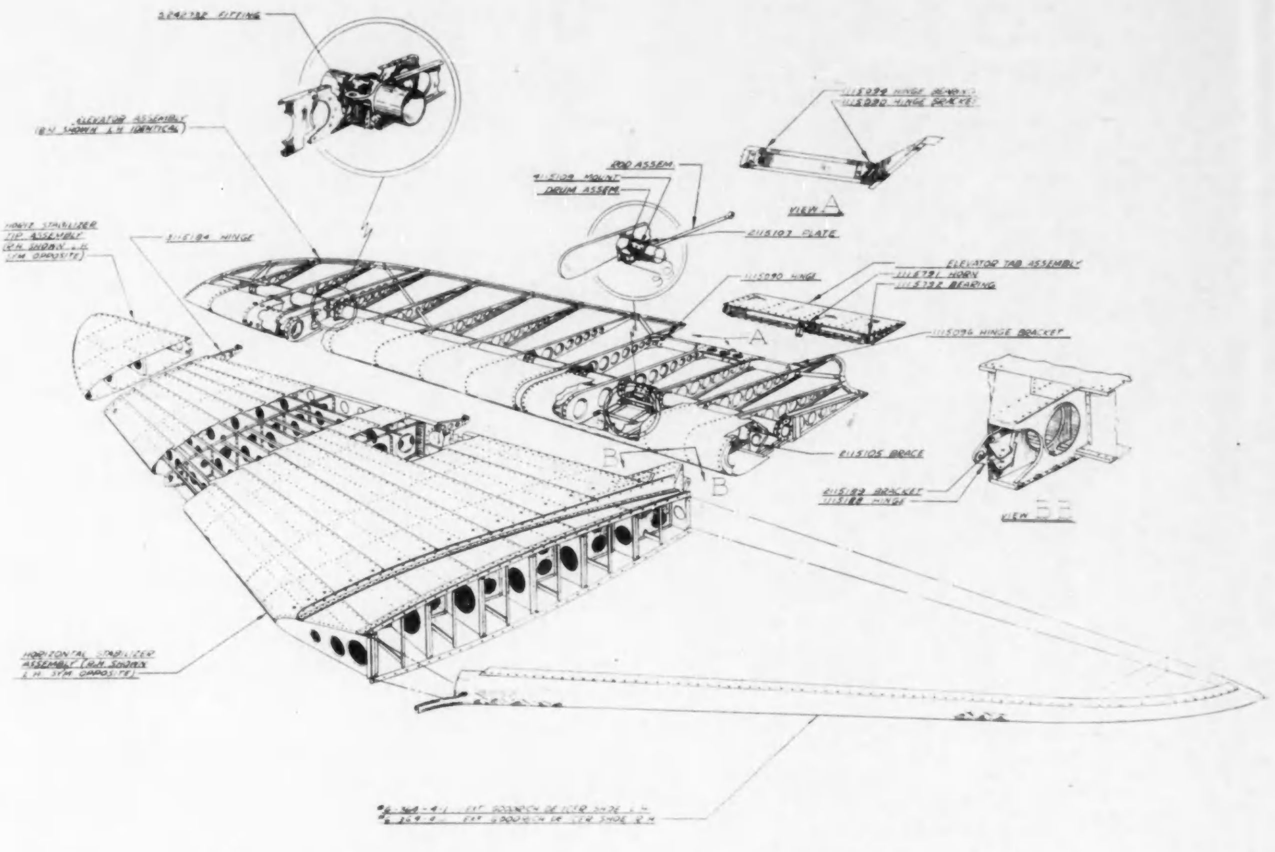
■ Phase I

The basic design for an airplane evolves from a specification submitted for competition by the Army or Navy and brought into being on paper by the Preliminary Design Department, ably assisted by the various specialty engineers.

At this stage the Production Illustration Department studies the new design and breaks it down into manufacturing units after consultation with the Engineering Production Design Group and Tool Design Department. Ultimately it issues a Production Breakdown Diagram, drawing in perspective as shown in Fig. 1. This diagram is accepted as the basic breakdown to be used by the Engineering Division for structural design and functional installation design, and by the Tool Design Department in like manner for the design of major jigs, tools, and fixtures. The Factory Planning and Shop Fabricating and Assembly Line Development Groups assigned to the



■ Fig. 3 – Working load graph between preliminary design stage and actual freezing of production techniques on final assembly line



■ Fig. 4 – Structural unit breakdown illustration – horizontal stabilizer, elevator, and tab

project refer to it constantly in performing their functions. So also, the Lofting, Production Planning and Estimating Departments find it to be of value in planning their work, as do some other departments to a lesser degree.

During the period of design in the Engineering Department, perspective illustrations serve the following purposes:

(A) Coordinate work between design groups by furnishing a composite picture of the activities of all groups as their work progresses.

(B) Promote more rapid design decision, especially in the earlier engineering stages of the project, by condensing the information obtained from partially completed layouts and thereby drawing attention to specification items vitally affecting the work in progress.

(C) Aid the "mockup men" in visualizing both the engineering requirements and new ideas.

(D) Suggest unitization of equipment to be installed.

(E) Save layout time by furnishing each group leader with a "picturized walking mockup."

As most projects these days fall under the Army or Navy "restricted" classification, special care is taken to restrict the distribution of prints and these are issued only to the group leaders of Hydraulics, Controls, Armament, Fuselage, Wings, Electrical, Power Plant, and Mockup, and recipients of these illustrations are requested to keep them unknown to those not working on the project, and must sign for their receipt.

Perspective illustrations are kept up to date by the Production Illustration Group. Since they represent incomplete production drawings at this stage, each print is dated and the project engineer's signature is secured before distribution. Due to the military restrictions mentioned, illustrations of this phase of the work cannot be reproduced for this paper.

■ Phase 2

The next step is the "Structural Unit Breakdown Illustration" (see Fig. 4). Each structural unit assembly as shown on the production breakdown diagram (Fig. 1) is illustrated separately in true perspective and further broken down into manufacturing sub-assemblies. This breakdown is made in consultation with the Engineering Production Design Group and the Tooling Department. This work is done during the preliminary design stage. The sketches are used by the design group as a guide for structural and detail design and by the Tooling Department for the scheming of sub-assembly jigs, tools, and fixtures.

On completion of the engineering design, this structural unit illustration is brought up to date and issued to all departments concerned.

The Materiel Division is responsible for the distribution of copies to any subcontractors concerned. If, for example,

the inner wing is manufactured outside the Douglas plant, the subcontractor receives a structural unit breakdown illustration of the complete unit, plus separate illustrations of each functional installation, such as, "Engine Installation," "Fuel and Oil Lines," "Hydraulic," "Electrical," "Landing Gear" as shown in Fig. 5. Subcontractors find these breakdown illustrations of tremendous value in planning, supervising, and facilitating construction, particularly in cases where they have had no previous experience in aircraft fabrication.

The next step is the making of "Functional Installation Illustrations" (see Fig. 6). Concurrently with the preparation of engineering layouts, accurate perspective drawings are made of all functional installations such as "Flight Controls," "Hydraulic System," "Fuel and Oil Systems with their Controls," and so on. This practice enables all concerned to obtain a clear and concise picture of each installation as it is assembled in the ship.

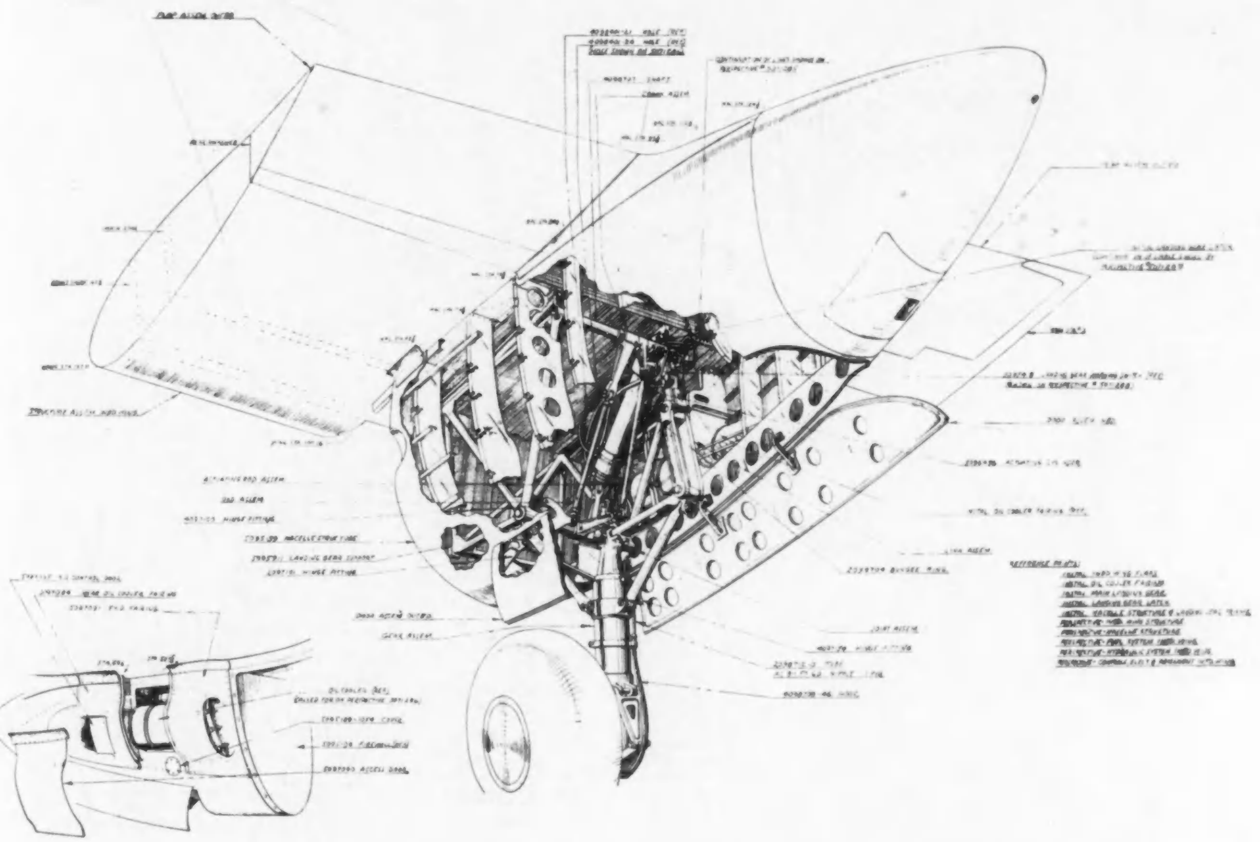
These drawings are used by the various engineering groups, by the Handbooks Department for inclusion in their Service and Instruction Manuals, by the Materiel Division for ordering purposes, by the shop coordinating supervisors and their development men for studying the assembly-line breakdown, and generally throughout the factory for instructional purposes. Instructors in the company's Educational Department find these diagrams especially effective in the training of new company employees. However, the intimate nature of the information which they contain considerably restricts their use for outside educational activities.

■ Phase 3

This phase starts when enough engineering and major tooling information is available for a profitable study of fabricating and assembly operations to be made by the Manufacturing Development Group. This group is composed of the coordinating supervisor, supervisor, and specialist leadmen who will carry the airplane through to completion. It is their duty to estimate, from information given on delivery dates, the number of positions for the fabrication, sub-assembly and final assembly, together with floor space and storage space necessary to complete the job in a given time.

Each final assembly position is based on an elapsed time sequence which may be applied to a track line operated by a link-belt system, moving either continuously or periodically.

The Production Planning Department becomes fundamentally involved at this juncture. Close coordination is necessary to insure the even flow of sub-assemblies and parts from their respective departments at a steady rate, along lines that are short, motile and clearly defined, so as to arrive at their predetermined positions in the shop at



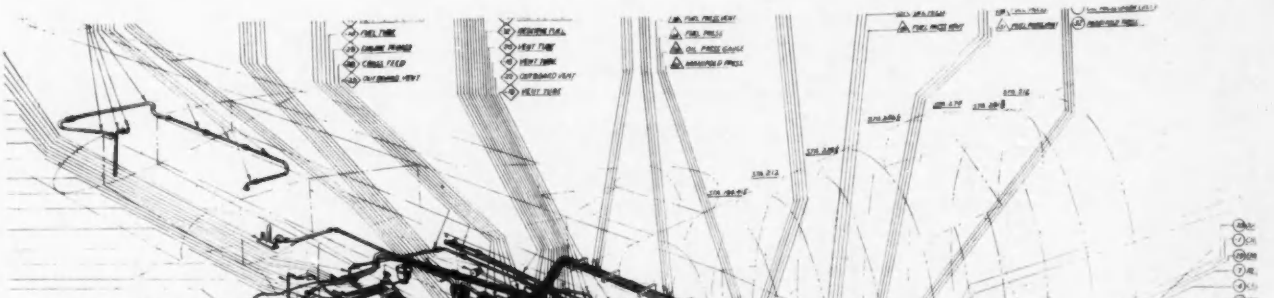
■ Fig. 5 – Structural unit breakdown illustration of functional installation – inboard wing and nacelle joined and landing gear

the times planned. For example, if the final assembly line were planned to move at 2-hr intervals, it would be necessary to consider every operation no matter how small, and to balance most carefully the work to be done by each mechanic in each stage for a 2-hr period. The work is so arranged that the mechanics do not interfere with each other. At the same time, the maximum amount of work is assigned to each position, because of unnecessary or a not fully productive position means a waste of valuable floor space.

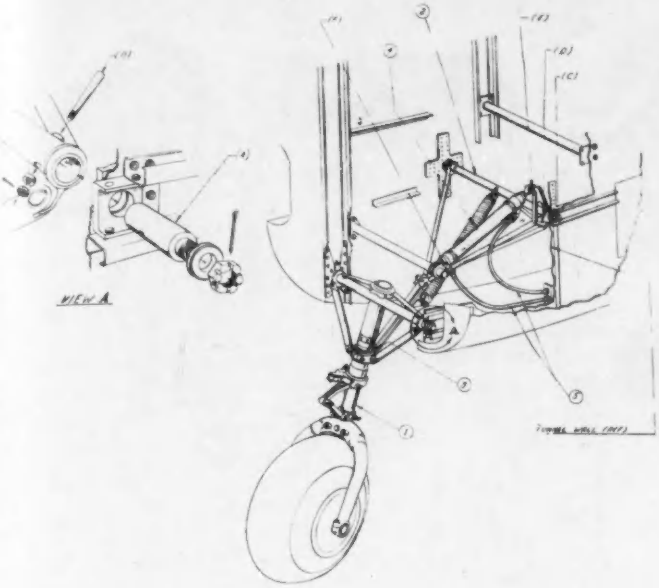
However this problem is not the responsibility of the Production Illustration Group. Its primary duty is to supply the illustrations necessary to give each of these other departments and groups a clear and concise picture of the work to be done. Here then, is where the Production Breakdown Diagram, the complete set of Structural Unit Breakdowns, and the Functional Installation Illustrations are again extensively used to help clarify and supply a coordinated picture of the engineering blueprints.

To help the development men estimate the allocation of work necessary in each zone for each position, Key Sheets are made. These Key Sheets are skeleton perspective sketches of the fabricated structure in each zone, showing frames, angles, and attaching brackets for receiving the several installations. Black and white prints covering each position on the line are issued to the development men. These black and white prints are easy to sketch on and it is their job to mark in the work to be done on their particular installation, indicating the approximate location of each part or sub-assembly and giving a list of tools necessary to do the job with the numerical sequence of the operations necessary. Similarly, a list of the nuts, bolts, screws, clips, and so on, required to attach each part is included.

When all the key sheets are marked they are used as: (1) a means of checking against overloading, or overcrowding any one position or installation zone; (2) a means of checking the location of various installations in order to prevent interference with other parts; (3) a means to aid the routing of parts or sub-assemblies to the production line by simplifying the determinative installation position and zone; and (4) a means of quickly locating installations



■ Fig. 6 – Example of functional installation illustration



JOB TICKET NO. 1812-22 MODEL A-20B

TOOL REQUIREMENTS

1/2" DRIVE RATCHET	1 PAIR DIAGONALS
1/2" DRIVE 1" SOCKET	1 PAIR PLIERS
3/8" DRIVE 1/2" SOCKET	1 BRASS BAR 10" LONG
3/8" DRIVE RATCHET	1 12OZ. BALL PEEN HAMMER
1 LARGE SCREWDRIVER	

PROCEDURE

OPER. NO.

1. INSTALL STRUT ASSEMBLY
APPLY A THIN COAT OF PARKER THREAD LUBE TO PIN (A) BEFORE INSTALLING. INSTALL AS SHOWN IN DETAIL "A".

(A)	1098970	PIN	8 REQ.
	1098967	WASHER	2 REQ.
	AN380-4-4	COTTER	2 REQ.
	AN320-10	NUT	2 REQ.
(B)	AC386-4+13	TAPER PIN	2 REQ.
	AN380-2-2	COTTER	2 REQ.
	AN975-5	WASHER	2 REQ.
	AN320-5	NUT	2 REQ.

2. INSTALL MECHANISM ASSEMBLY
USE SAME PROCEDURE AS IN STEP NO. 1.

(C)	1090965	PIN	2 REQ.

3. CONNECT MECHANISM ASSEMBLY TO STRUT ASSEMBLY

	AN29-36	BOLT	1 REQ.
	S-102661489-108	SPACER	1 REQ.
	AN320-9	NUT	1 REQ.
	AN960-916	WASHER	1 REQ.
	AN380-4-4	COTTER	1 REQ.

4. INSTALL CYLINDER ASSEMBLY

(E)	AN6-15	BOLT	1 REQ.
	S105960-8-375-624-749	SPACER	1 REQ.
	AN310-6	NUT	1 REQ.
	AN960-616	WASHER	1 REQ.
	AN380-3-3	COTTER	1 REQ.
(F)	AN6-14	BOLT	1 REQ.
	S105960-8-375-624-749	SPACER	1 REQ.
	AN310-6	NUT	1 REQ.
	AN960-616	WASHER	1 REQ.
	AN380-3-3	COTTER	1 REQ.

5. CONNECT HOSE #4098400-21 & -18 TO CYLINDER
6. TIGHTEN AND SAFETY ALL NUTS AND BOLTS

■ Fig. 7 – Illustrated job ticket – Job No. 1812-22 – nose-wheel installation

or assemblies on the production line, or any changes therein.

When all key sheets are assembled and coordinated, job tickets are issued by the Planning Department with a job number. This job number first gives the position number on the line, -18, its installation code number, -12 (means hydraulic assembly units), and its zone number, -22 (which is indicated on Fig. 2 as the zone under the pilot's floor). This gives a "Job Number," 1812-22, as shown in Fig. 7.

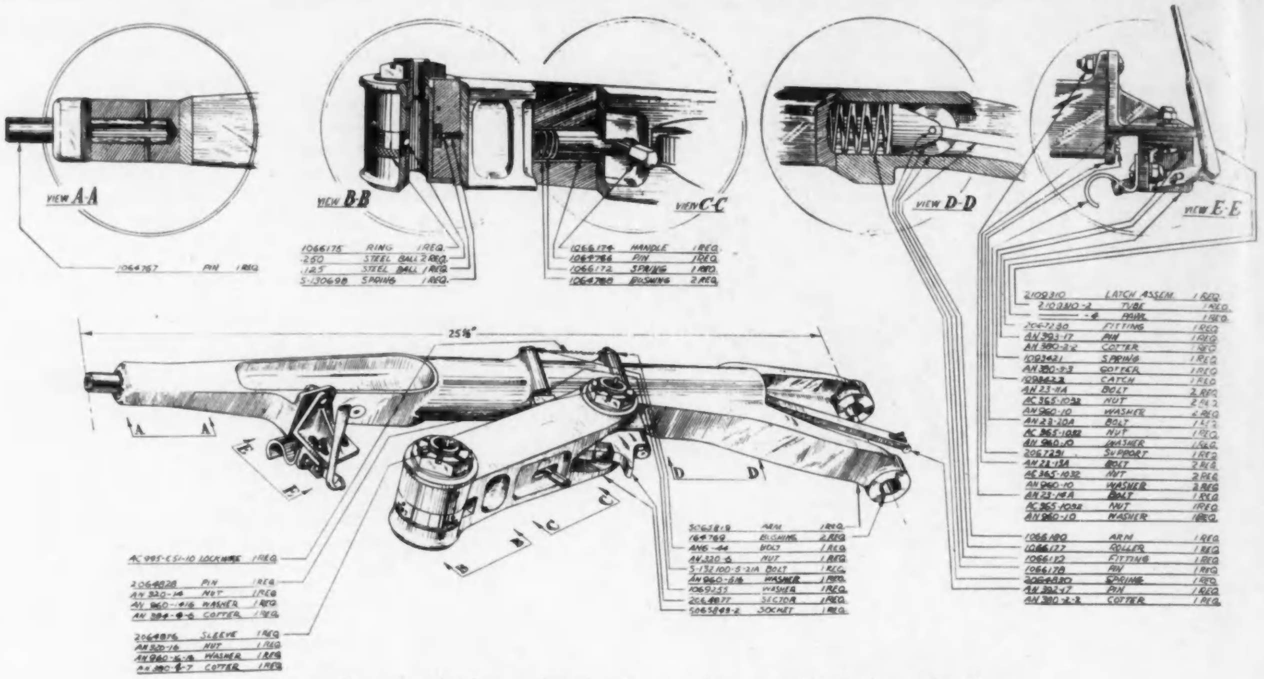
The Production Illustration Department then takes each of these job tickets in their order of priority and, from their accompanying marked-up key sheets makes the Illustrated Job Tickets, of which Fig. 7 is an example. Fig. 7 was picked at random and represents the final installation of the nose wheel, being a 2-hr elapsed-time job ticket. On the right-hand side is a list giving the tools necessary for the job followed by a listing of each operation in numerical sequence with a corresponding number on the sketch. It will also be observed that Operation No. 1, for example, is subdivided into two sub-operations (A) and (B) giving an enlarged view of the parts and their pictorial sequence of assembly. If the instructions are carefully followed, it is impossible for the mechanic to make an error.

Sub-Assembly Sketches are made concurrently with the illustrated job tickets. These drawings, of which Fig. 8 is an example, are made after consultation with the Production Planning Department and are used as an illustrated guide for bench assembly or sub-assemblies. They provide a complete list of parts, plus the necessary instructions to complete the jobs ready for storage in their respective bins until they are transported to and issued at their appointed place on the assembly line as called for on the Illustrated Job Tickets.

These drawings, together with the illustrated job tickets, are issued by the Production Planning Department to Materiel Control which uses them as a basis for the routing of sub-assemblies, material and parts to the Assembly Line Stock Rooms which use them as a guide for the issuing of parts in accordance with the job tickets, to the Time Standards Department, and to the Inspection Department where they are used to plan the work of individual inspectors assigned to the assembly line.

■ Phase 4

It is expected, of course, that changes and slight improvements will be made during the initial period of operation of an assembly line on which new designs are being fabricated. Allowance is made for these adjustments to as great a degree as 30%, as indicated by the curve marked "Changes" on Fig. 3. For example, the work done by each individual is adjusted so as to fit more accurately into the assignment of work to be done at each position and into



■ Fig. 8 – Example of sub-assembly sketch – mounting assembly for lower flexible gun

the elapsed time schedules established for the line as a whole.

This brief outline summarizes some of the more important uses to which Production Breakdown Illustrations are put. Innumerable other instances of their value as a factor in the coordination and solution of complex production problems could be cited.

The ancient device of portraying to the eye the ideas behind new and sometimes startling concepts of the human mind, so effectively used by Leonardo da Vinci, has been enlisted to serve the cause of Democracy.

Recent Progress in Ocean Air Transport

DESPITE the problems during the past three difficult years in which the transatlantic route has been in operation, some progress can be assumed in the developing art of ocean air transport. That our operating organization has succeeded in maintaining a reasonable record of service and, to date, an unblemished record for safety, is a source of gratification. The methods developed to achieve this record represent our contributions to air transport. They are, to a large extent, original with Pan American Airways.

First came the development of the multiple flight crew, through the specialized organization and training of which it has been possible to conduct transport flights as long as 24 hr in duration. Today's multiple flight crew is composed of a captain and his crew of ten. Directly responsible to the captain are five officers qualified as specialists in the primary departments of piloting, navigation, engineering, communications, and passenger service. Each of these has a qualified assistant capable of relieving him for rest periods. The crew of 11 is as highly organized as any which ever operated a destroyer or a submarine.

Second came the development of accessibility of powerplants in flight which is of inestimable value. The flight deck of the Boeing B-314 is ample for the entire flight crew and is located above and completely separated from the passenger compartment. By means of a companionway leading through the wing, the flight engineer makes periodic visible inspection of the mechanical functioning of the engine and its electrical, fuel, and oil system. He also has available for inspection all main fuel lines, all control lines, and all interior elements of the wing. During the first two years of operation with this type of Clipper, there were 431 cases of repairs made in flight. In 64 of these cases, had the engine not been accessible, the plane would have been required to return to its nearest base with only three of its engines in operation. The safety advantages are obvious. The saving of each such flight completed on schedule is in itself an economic factor of importance.

The third method developed by the company was scientific control of flight.

Each flight is carefully planned. To execute the plan accurately, it is necessary to have in hand an easily usable yardstick of performance. We have developed such a yardstick which bears the very scientific title of a "Howgozit Curve." Actually the "Howgozit Curve" is made up of 5 curves: (1) normal four-engine fuel consumption; (2) gallons versus hours of flying; (3) hours versus miles; (4) miles versus gallons for three engines ahead conditions; (5) miles versus gallons for three engines return conditions. And on this chart is also the "point of no return."

The three methods just outlined – multiple flight crews, accessibility of powerplants in flight, and the scientific control of flight – have been most effective.

Excerpts from the paper: "International Air Transportation," by H. W. Peterson, district sales manager, Pan American Airways, presented at a meeting of the Southern California Section of the Society, Los Angeles, June 5, 1942.

The Testing of HEAVY-DUTY MOTOR OILS

by HARRY C. MOUGEY[1] and JOSEPH A. MOLLER[2]

THE work described in this paper was authorized by Subdivision B of the Lubricants Division, SAE Standards Committee, with the view toward the establishment of a test procedure by which so-called heavy-duty motor oils might be evaluated. A further object of this work, carried out by two committees of Subdivision B – an Automotive Committee and a Petroleum Committee, under the chairmanship of Authors Mougey and Moller respectively, was the establishing of limits, in so far as engine designs are concerned, for the stresses which they impose on lubricating oils.

It was necessary to establish reproducible test conditions that were sufficiently severe so that the properties of the oils could be evaluated in a reasonable period of time, and yet such that the test conditions should not be so severe as to impose unreasonable demands on the lubricating oil.

This paper deals with the results obtained from the participating laboratories and the correlation of these tests with performance in service.

The test procedures recommended in this paper have been issued in ASTM standard form, for information.

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THE AUTHORS: HARRY C. MOUGEY (M '20), since October, 1939, technical director of the Research Laboratories Division, General Motors Corp., graduated from Ohio State University in 1911 with the degree of B. Sc. in chemical engineering. In 1935 he received the degree of chemical engineer. After graduation in 1911 he was research chemist with the Lowe Brothers Co. in Dayton, joining the staff of the Dayton Metal Products Co. in 1917. When this company in 1920 became a part of GMC, Mr. Mougey was made head of the General Chemistry Department of the Research Laboratories. Later he was promoted to assistant technical director, and then to technical director – his present position. During World War I, he was associated with Dayton-Wright Airplane Co., in charge of physical and chemical tests. Mr. Mougey has been active on many SAE Research and Standards Committees. JOSEPH A. MOLLER (M '35), has been chief products engineer of The Pure Oil Co. in Chicago, and is now a lieutenant-colonel in the Army Air Forces.

SOME two or three years ago, the petroleum and engine manufacturers were having spirited discussions and disputes regarding lubricating and engine operating difficulties. There was a tendency for the engine manufacturers to blame the oil for all of their troubles and for the refiners to claim that the engine designs were at fault.

Numerous examples could be given which seemed to prove the arguments of either group. For example, a certain engine design might operate at high piston temperatures, a hot ring-belt condition which would result in stuck rings, under severe operating conditions. This condition could have been overcome by lowering the engine output, by using an oil capable of withstanding these temperatures, or by changing the basic design to effect a cooler ring-belt area. In the same way, another engine, under certain operating conditions, might have trouble because of oxidation of the oil due to high, or hot, sump temperatures which would cause excessive sludges, with their attendant difficulties, or excessive and rapid bearing corrosion, with the inevitable mechanical failure of the engine. Here again, ultimate failure could have been overcome by lowering engine output, or by using an oil having sludge- and acid-resisting tendencies for the operating conditions encountered or by changing the basic engine design so that cooler sumps could be obtained, still maintaining the high output desired.

However, the obvious difficulty was simply that neither the petroleum industry nor the engine manufacturers had any means of measuring, or determining, what were the suitable composite design limits of engines in so far as lubrication was concerned. Further, it is obvious that, if commercial oils were available, which would withstand certain operating conditions, it was the duty of the engine designer to utilize these properties to the best design and operating advantage possible, and to provide, of course, for some factor of safety.

To sum up the situation as it stood some two years ago, the engine manufacturers were right in that the oil was failing to do a satisfactory job in certain engines which were being operated in severe service. On the other hand, the petroleum manufacturers were equally right in that engines were being designed which could be so operated as to place greater stresses on one of its design parts[3], the oil, than that part could withstand successfully.

Certain of the engine manufacturers, recognizing this problem rather early, had devised engine tests which, if

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 16, 1942.]

[1] Technical director, Research Laboratories Division, General Motors Corp.

[2] Chief products engineer, The Pure Oil Co.; at present lieutenant-colonel in the Army Air Forces.

[3] The fact that the lubricating oil is considered as a design part of an engine has been recognized for some time. See "Heavy-Duty Oils," by Harry C. Mougey, presented at the Annual Chemical Engineering Homecoming Program, Ohio State University, Columbus, O., Nov. 15, 1941.

successfully withstood by lubricating oils, would insure satisfactory field service. However, each manufacturer had his own test, or series of tests, on his own engine, and it did not follow that an oil which would pass one engine manufacturer's tests would, therefore, pass another's. The situation for the future appeared even more gloomy in that it seemed assured that, before long, almost every manufacturer of high-output internal-combustion engines would have to run tests to determine what oils would be satisfactory for his engines in the field.

Such a situation, if expanded to its logical conclusion, therefore, would involve an expensive and continuous experimental and test program for each refiner and for each engine manufacturer.

When all the factors which contributed to this situation had been considered and discussed, there were but two solutions which could be evolved. The first solution was to do nothing about the situation at all, but to let it work itself out along the then-present lines. Each engine manufacturer, as he ran into field trouble with his engines, could set up his own engine test program to determine what oils would function satisfactorily in his engines. To be fair, he would have to test all oils submitted by all the refiners. The refiners would also have to institute similar programs for the development of an oil, or oils, which could pass such engine manufacturer's series of tests. In addition, he would then have the obligation of stocking all of these approved oils in all of the proper grades in the field, so that each engine would be serviced with only the oil designated by the engine manufacturer. In the long run, whether these tests were paid for by the refiner or engine manufacturer, the cost of all of the programs must be reflected ultimately to the engine user, in higher engine price, or in an increased oil price, or both.

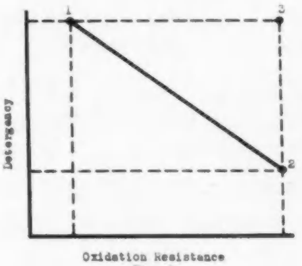


Fig. 1

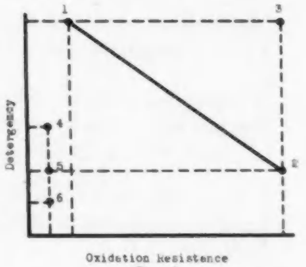


Fig. 2

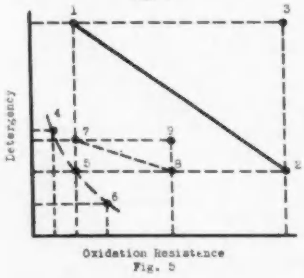


Fig. 3

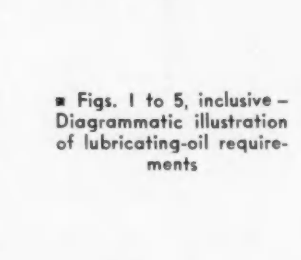


Fig. 4

Fig. 5

Figs. 1 to 5, inclusive – Diagrammatic illustration of lubricating-oil requirements

The second and obvious solution would be, in the light of the knowledge obtained, to develop a method of testing by which the various factors of engine and oil design could be evaluated and described in language and in terms understandable by, and between, the engine and petroleum manufacturers.

However, in arriving at such a solution, it was not only desirable, but extremely necessary, that present-day field requirements be considered. In other words, if such test methods, for both immediate and future commercial use, were to be devised for the evaluating of oils, they should be such that present-day engines could and would be operated satisfactorily on oils meeting these test conditions. Again – in fairness to both industries if such a test method could be developed satisfactorily – if the refiners would agree to make these oils commercially available in the proper grades, on the one hand; then, on the other, the engine manufacturers, having available such design information for the first time, must also agree to use these data as an integral part of their engine design.

Obviously, progress will always have to be made. As the knowledge of the alloying of petroleum products is increased, the limiting descriptive factors in the test method procedure can be altered, increased, or decreased by one means or another, as may be deemed advisable from time to time. The point is that a usable, understandable, reproducible, and practical description and specification of lubricating oils must first be agreed to by all those concerned with the design and operation of engines under all types of service conditions.

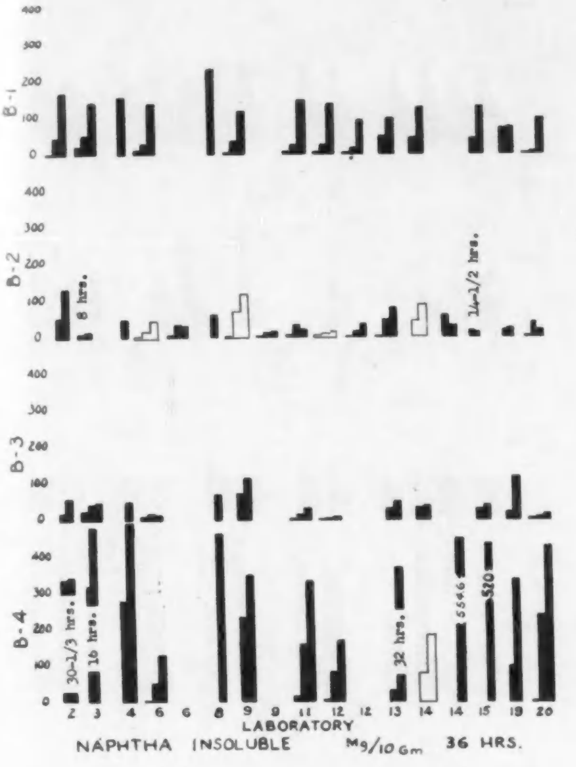
Further, since it is recognized that progress must be made in both engine and oil design; therefore, the second and equally as important step is that, once such a test procedure and description has been evolved, changes such as must necessarily be demanded by progress, shall be made only with the consent and approval of those affected by such changes.

However, it should be recognized by both industries that "surprise" increases by either industry, or by individual concerns in those industries, would without doubt be commercial mistakes which would react to the disadvantage of the group springing such a surprise.

Requirements for Test Procedures

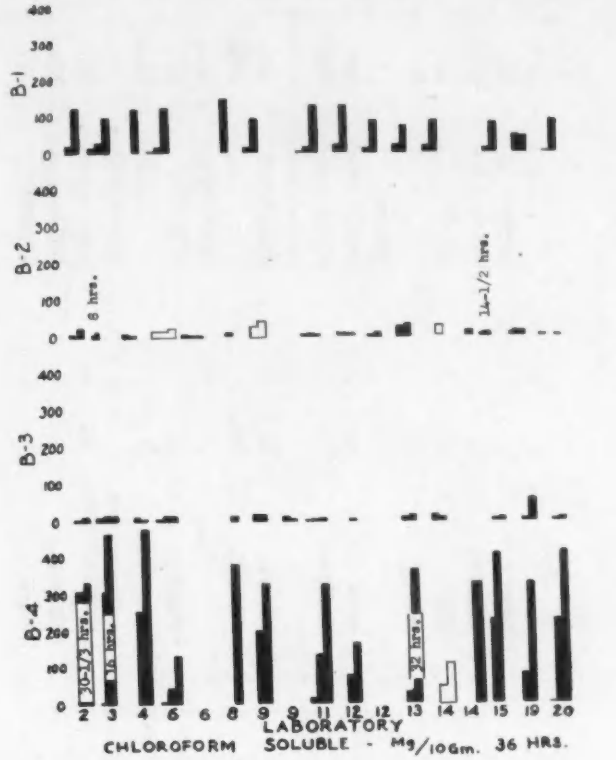
With these objectives in mind, a joint meeting of the two SAE committees (Automotive and Petroleum Committees, Subdivision B, Lubricants Division, SAE Standards Committee) was held early in 1941 to work out such a procedure. It was indeed fortunate that sufficient work had been done so that most of the desirable and the undesirable properties of heavy-duty motor oils were quite generally understood. The only problem was, therefore, to define, describe, and compare these properties by means of the same common yardstick.

It was conceded quite generally that the Caterpillar series of tests, as then outlined, quite adequately defined and described the detergent and dispersive properties of the oil; that the General Motors Series 71, 500-hr test defined and described to some degree the combination of both the detergent and dispersive properties, together with the oxidation characteristics of the oil; and that the 36-Hr



White bars - not in accord with test procedure in all respects.

■ Fig. 6 – **Naphtha insoluble reported by participating laboratories for B-1, B-2, B-3, and B-4 oils**



White bars - not in accord with test procedure in all respects.

■ Fig. 7 – **Chloroform soluble reported by participating laboratories for B-1, B-2, B-3, and B-4 oils**

Oxidation Test Procedure (Table 5, Appendix) placed the greatest emphasis on the oxidation characteristics of the oil.

In other words, the Caterpillar series tested primarily the detergent and dispersive characteristics of the oil, while the 36-hr Oxidation Procedure tested the oxidation characteristics.

Fig. 1 illustrates this point. Let us assume that point 1 represents the Caterpillar series of tests and, while the value of required oxidation resistance is unknown, it is known that this value is low. By the same token, the detergency is high.

The assumption may also be made that point 2 represents the 36-hr Oxidation Test which is high in oxidation-resistance value, though the required detergency value, while unknown, is known to be low.

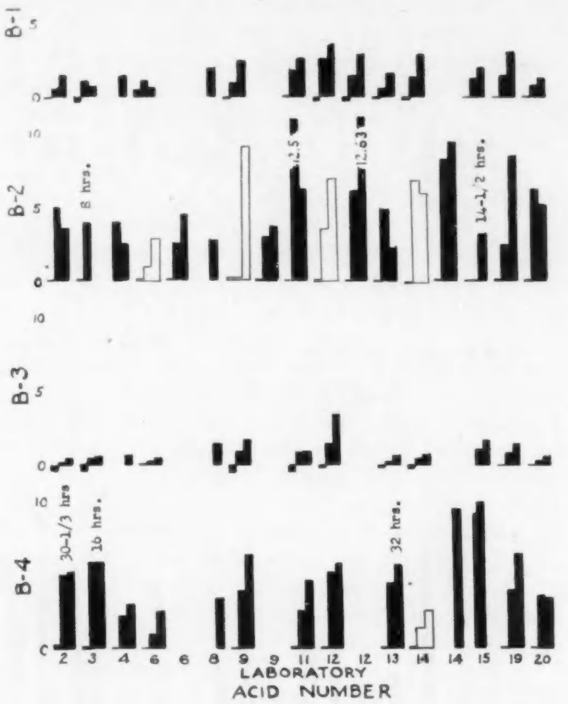
Thus, diagrammatically, the two "ends" of the lubricating-oil requirement scale are shown. Obviously, an oil which successfully passes tests 1 and 2, would pass some third test, yet to be devised, which could be plotted as point 3. To sum up, points 1 and 2 are representative of two known and understood test procedures.

Fig. 2 again shows diagrammatically the same three points which were illustrated in Fig. 1. However, in addition to the points already described, additional points have been added. Points 4, 5 and 6 are plotted diagrammatically representing "straight" mineral oils of three basic crudes. These points may be arrived at by either of two methods. The first method would be to decrease the severity of the tests, which give point 1, until the oil "passes" the test. The second method would be to plot the percentage rating of the engine tests using point 1 procedure as 100%.

Fig. 3 is arrived at in the same manner as is Fig. 2 except, in this case, these same three oils are evaluated by either, or both, of the methods just described, but using the test procedure outlined for obtaining point 2.

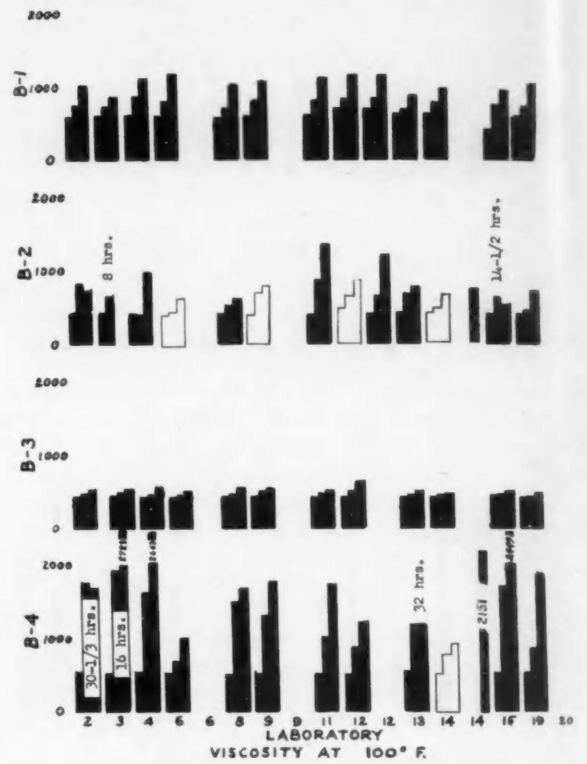
Fig. 4 is a diagrammatic plot showing the combination plotting of the relative values obtained for these "straight" mineral oils. In other words, these three points represent diagrammatically what straight mineral oils of three basic crudes will do when these two tests are used for evaluating them.

It must always be borne in mind that the severity of any set of field operating conditions is not only one of temperature, whether ring belt or sump, or any combination of these two, but that *time* must also be considered. In other words, the tests under discussion (with, perhaps, the exception of the Caterpillar Scratch Test) deal only with the oxidation resistance, and the detergent and dispersive ability of the oil. Obviously, oxidation of oils is a function of *time* times temperature. By the same token, though perhaps not quite so obviously, the amount of material which the detergent properties of the oil must handle depends both upon the amount already present in the oil at any particular time, and the rate of formation of newly oxidized material from the lubricating oil, plus the amount of material in, and the rate of, the blowby gases. In other words, since the strength of oils to resist physical and



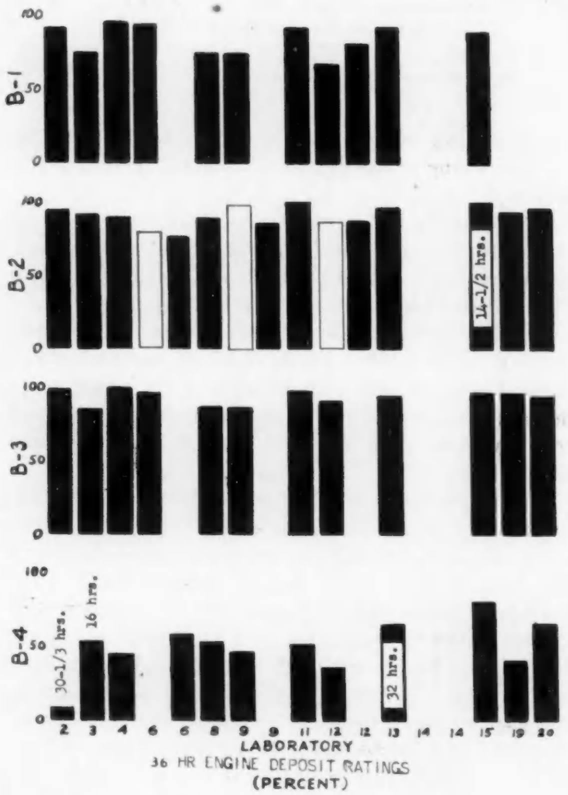
White bars - not in accord with test procedure in all respects.

■ Fig. 8 – Acid number reported by participating laboratories for B-1, B-2, B-3, and B-4 oils



White bars - not in accord with test procedure in all respects.

■ Fig. 9 – Viscosity reported by participating laboratories for B-1, B-2, B-3, and B-4 oils



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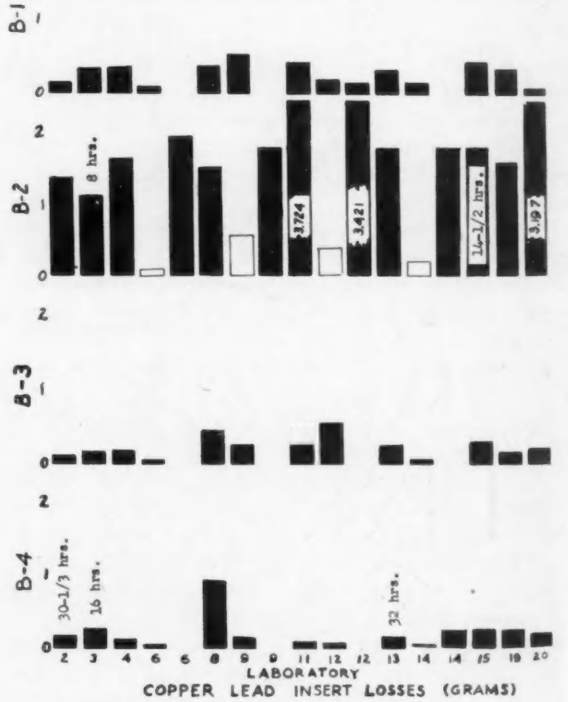
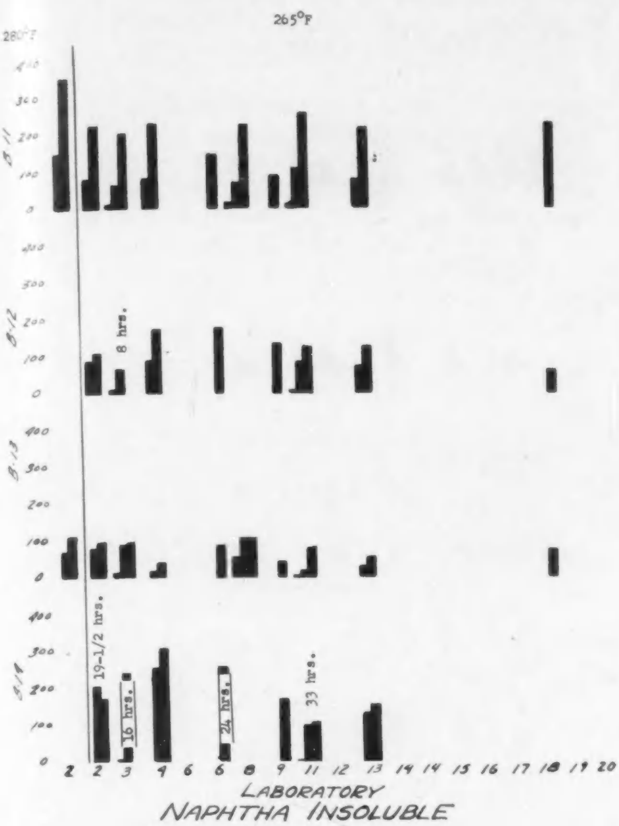
■ Fig. 10 – Engine deposit ratings (36-hr) reported by participating laboratories for B-1, B-2, B-3, and B-4 oils

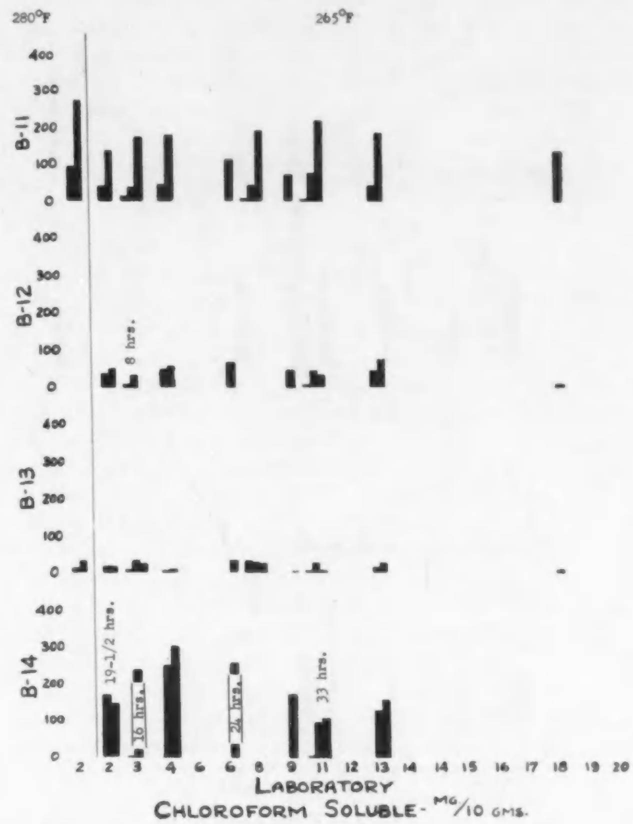


White bars - not in accord with test procedure in all respects.

■ Fig. 11 – Copper-lead insert losses reported by participating laboratories for B-1, B-2, B-3, and B-4 oils



■ Fig. 12 – Naphtha insoluble reported by participating laboratories for B-11, B-12, B-13, and B-14 (SAE 10) oils



■ Fig. 13 – Chloroform soluble reported by participating laboratories for B-11, B-12, B-13, and B-14 oils

chemical changes is basically a function of *time* times temperature, the engine manufacturer must declare himself regarding the length of time, either in miles or hours, after which his engine users must drain the oil. This factor, then, must constitute one of his basic test conditions.

Fig. 5 is Fig. 4 repeated, but in addition points 7, 8, and 9 are diagrammatically plotted to represent the severity of a set of field operating conditions (including declared drain periods), as determined by the reference oils for any particular engine. Obviously, some engines under certain operating conditions will stress one characteristic – for example detergency – over another characteristic – oxidation resistance – while the same, or another, engine under different operating conditions may reverse this order of importance of characteristics. Some so-called heavy-duty engines will, under some special field operating conditions, operate quite satisfactorily on straight mineral oils.

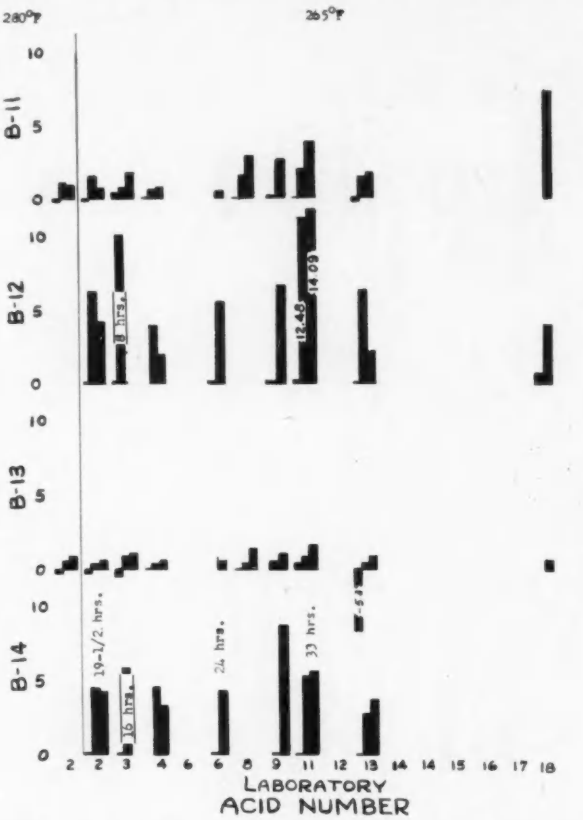
The point is that, if a yardstick consisting of the two test points as diagrammatically represented by point 1 and point 2, is acceptable to the engine manufacturers as an oil specification for heavy-duty service performance, and the petroleum refiners agree to the burden of manufacturing and supplying oils to meet this description, then must also the engine manufacturers agree that they assume the burden of providing an engine design for their manufacture which, for the operating conditions declared by them, will operate satisfactorily using oils meeting the same description.

Just so long as the sum total of the engine manufacturer's operating conditions are such that the tests, or test combinations, diagrammatically plotted by means of the reference oils on this graph, do not exceed the values on the ordinate and abscissa of the test conditions of point 1 and point 2 respectively, then that engine should be lubricated properly for the declared drain periods with these heavy-duty motor oils. On the other hand, should either or both of the engine test conditions, diagrammatically plotted, exceed the values set for point 1 and point 2, then the petroleum industry may properly and publicly call attention to such fact, and the engine manufacturer will, of necessity, have to take such steps as are necessary to remedy the situation for units already in the field, as well as for future production.

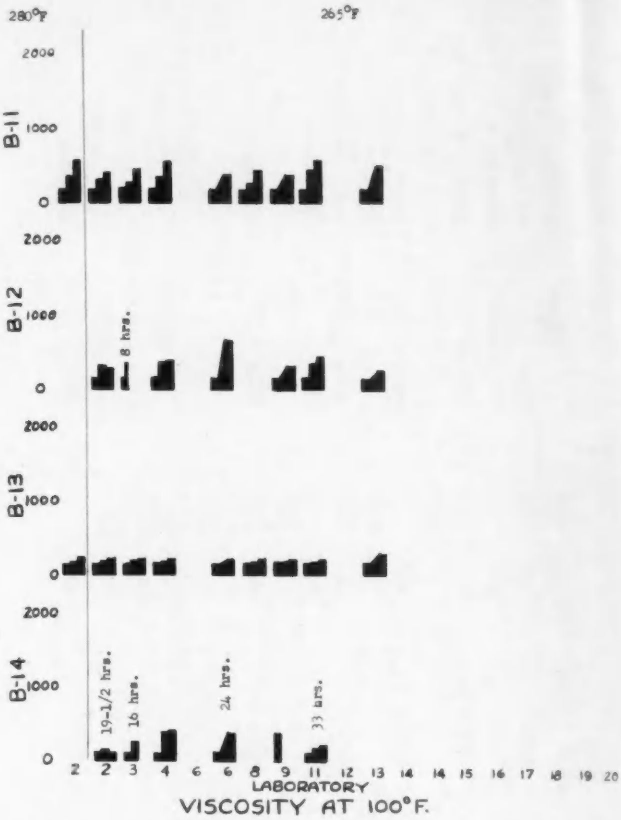
■ Results of Cooperating Laboratories

The results of the cooperating laboratory tests are, it is felt, quite remarkable in that but three test series have been run. The 67-hr, or first series, need not be reported here. While the first 36-hr, or second cooperative series, was reported in the report of Subdivision B, Lubricants Division, SAE Standards Committee, given in San Francisco[4], it is felt that a résumé of this work should be given.

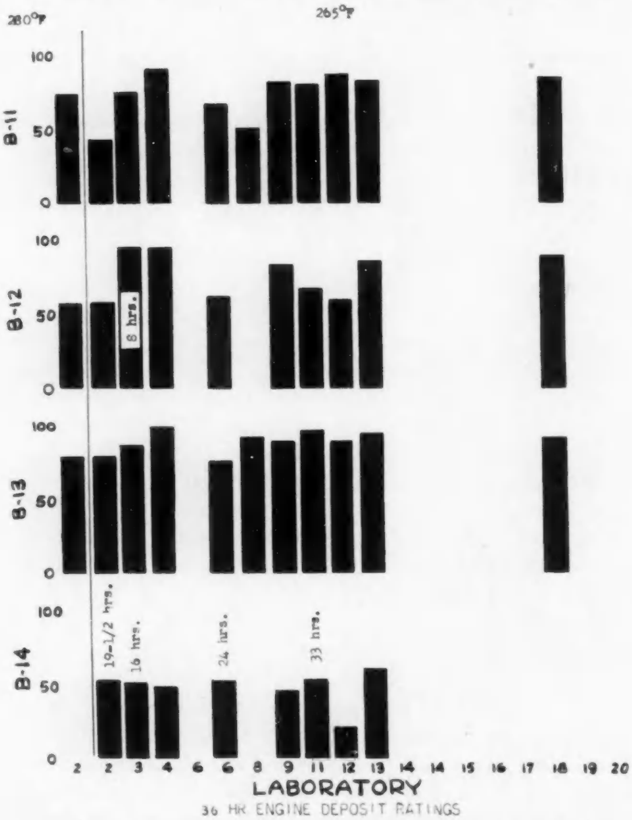
[4] See Proceedings of the 22nd Annual Meeting of the American Petroleum Institute, Nov. 3-7, 1941, San Francisco, Calif., Section III, Vol. 22, pp. 100-123: "The Testing of Heavy-Duty Motor Oils," by H. C. Mougey and J. A. Moller.



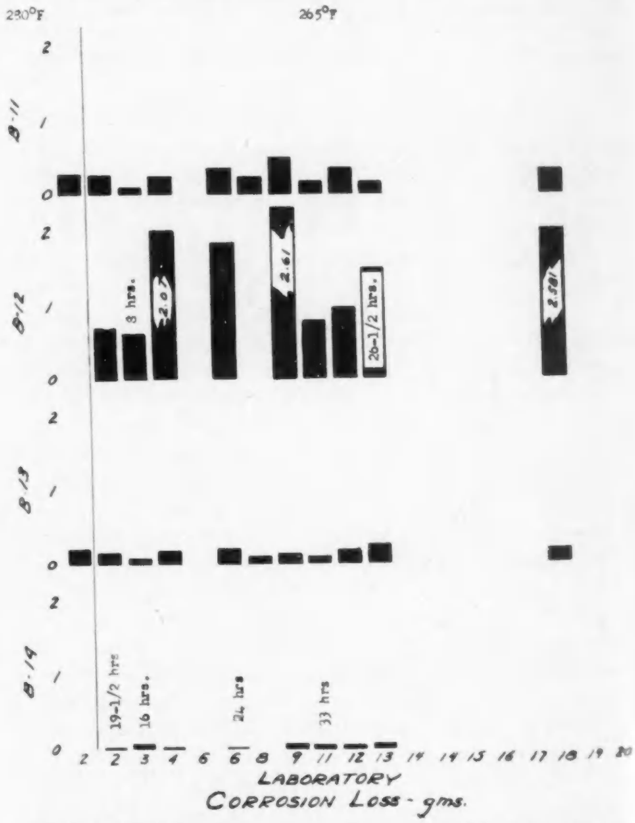
■ Fig. 14 – Acid number reported by participating laboratories for B-11, B-12, B-13, and B-14 oils



■ Fig. 15 – Viscosity reported by participating laboratories for B-11, B-12, B-13, and B-14 oils



■ Fig. 16 – Engine deposit ratings (36-hr) reported by participating laboratories for B-11, B-12, B-13, and B-14 oils



■ Fig. 17 – Relative bearing corrosion reported by participating laboratories for B-11, B-12, B-13, B-14 oils

The principal reason is that certain laboratories were somewhat out of line, and re-runs were made. Upon completion of the additional runs, the corrected data are herewith given.

Fig. 6 illustrates the corrected results as reported by participating laboratories for the B-1, B-2, B-3, and B-4 oils for naphtha insoluble. In this, and in the succeeding figures for these oils, the second bar for each laboratory indicates that some change in engine procedure or operating technique was necessary, and that, when these changes had been made, the data were then found to follow the same general pattern.

Fig. 7 is a similar slide showing chloroform soluble in mg per 10 g.

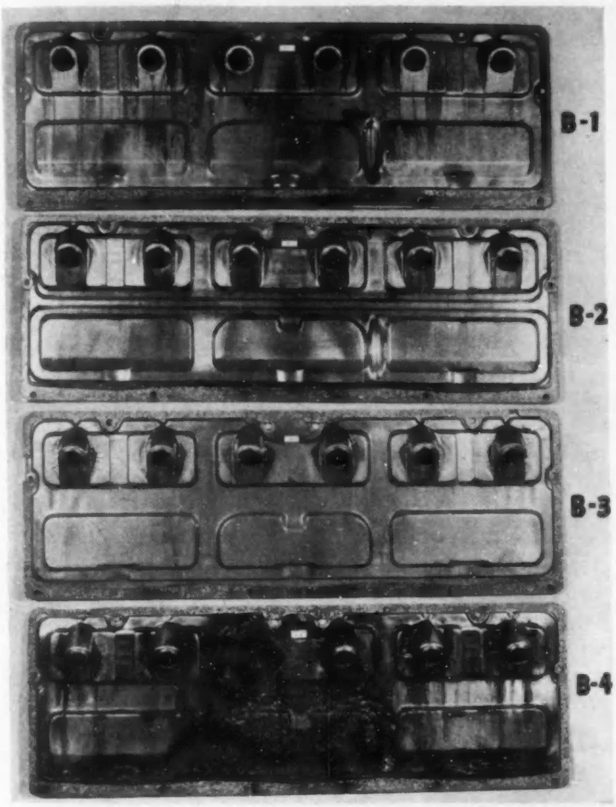


Fig. 8 illustrates the neutralization number of these oils as obtained by the laboratories. Note particularly, the relative acid number values for B-2 and B-4 oils. Note also the relative order of B-3 and B-1, in the order shown in one group; and B-2 and B-4 in another group.

Fig. 9 illustrates the viscosity of the oil as the test progressed. It should be noted that the steps in the bar show the viscosity of the oil new, at 16 hr and at the end of the test.

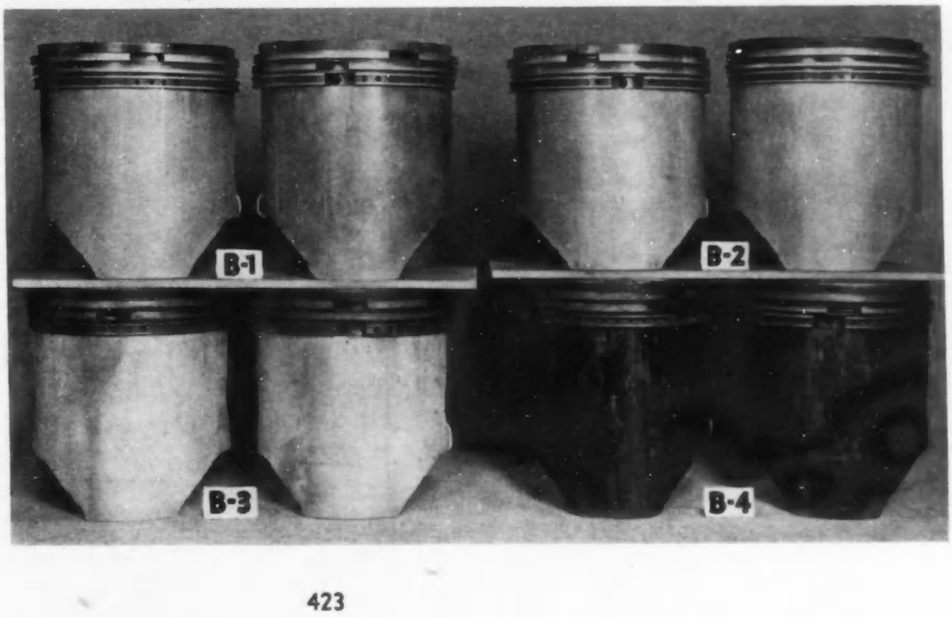
Fig. 10 is important compilation of data in that it clearly indicates the effect of the oxidation of the oil on the engine parts. The method of rating is discussed in the 36-hr Oxidation Test Procedure, and hence does not need to be discussed here.

The important observation to be made regarding these data as to the effect on engine parts due to oil oxidation is that, taking into consideration early cut-offs of B-2 runs due to excessive corrosion of its bearings, the oils group themselves, from the point of view of deposition only, in the order of B-3, B-1, and B-2 in one group and then B-4 in a class by itself. It is interesting to note, from an engine-deposition point of view, these results are in accord with field service data. It should be noted that B-2, however, will give "coffee-ground" results if used under too severe conditions for long periods of time.

Fig. 11 illustrates the relative corrosion of bearings as obtained on the reference oils. Recall the acid numbers of the oils, and note that the expected corrosion of B-2 and B-4 should be somewhere near the same value. However, these data indicate that this condition is not true. B-2 oil is highly corrosive, while B-4 oil is not.

The order in which the test oils are now placed, from the point of view of corrosion of the copper-lead bearings, might be read from these data as B-3, B-1, B-4 or, perhaps, B-4, B-3, B-1, in the order shown, in one group; and B-2 in another class by itself. Again the service data on these oils under heavy-duty operating conditions would confirm these results.

The foregoing oils, known as B-1, B-2, B-3 and B-4, are reference oils, and are of the SAE 30 viscosity classification. Because of the need of an SAE 10 viscosity classification oil in service, it was then decided to send out another series of oils of the SAE 10 range. This was done



■ Fig. 18 – Valve cover plates (above) and pistons from engines tested with B-1, B-2, B-3, and B-4 oils – parts from tests made by Laboratory No. 4

and the oils were called B-11, B-12, B-13 and B-14.

Fig. 12 illustrates the results of the cooperating laboratories for naphtha insoluble. The data for this figure, and for the following figures through Fig. 17, are to be found in Tables 1 to 4 in the Appendix.

Fig. 13 shows the relative results obtained by the laboratories for chloroform solubles.

Fig. 14 shows the relative neutralization numbers of the test oils. Note that B-13 and B-11 are in one group, and B-14 and B-12 in another group.

Fig. 15 illustrates the viscosity increase in the oil during the cooperative tests. It should be noted that the viscosity of B-14 was extremely low.

Fig. 16 gives the engine deposit ratings for the test oils. It should be noted that these ratings are of the engine "as opened." No account has been taken of the fact that, in some of the B-12 and B-14 tests, the run was terminated prior to 36 hr because of high acid number, bearing failure, or a stuck engine. Had some correction factor been applied, the results would have been more nearly in line. For example, if a run is terminated because of bearing failure in 18 hr, perhaps 18/36's of the reported rating should be taken as the final rating of the engine.

However, these ratings do group the oils from the point of view of deposition in the order of B-13, B-11, and B-12 in one group, and then B-14 in a group by itself.

Fig. 17 illustrates the relative bearing corrosion. Here again, some runs were cut off early. Also note that, from the neutralization number of B-14 (see Fig. 14), the expected corrosion of this oil would be about that of B-12. Perhaps the explanation is that the rapid rate of varnish formation may be sufficient to "protect" the bearings from the high acid content of the oil. Again perhaps, the acid formed by B-14 is not quite as corrosive as is the acid formed by B-12.

Whatever the explanation, the results, from the point of view of bearing corrosion, group themselves again into two groups – B-13, B-11 and B-14, or perhaps B-14, B-13 and B-11 on the one hand, and B-12 in a class by itself.

Summing up, it is found that B-12 fails because of bearing corrosion and that B-14 fails because of engine deposit. Field results would also indicate that B-12 and B-14 are unsatisfactory, while B-11 and B-13 are satisfactory.

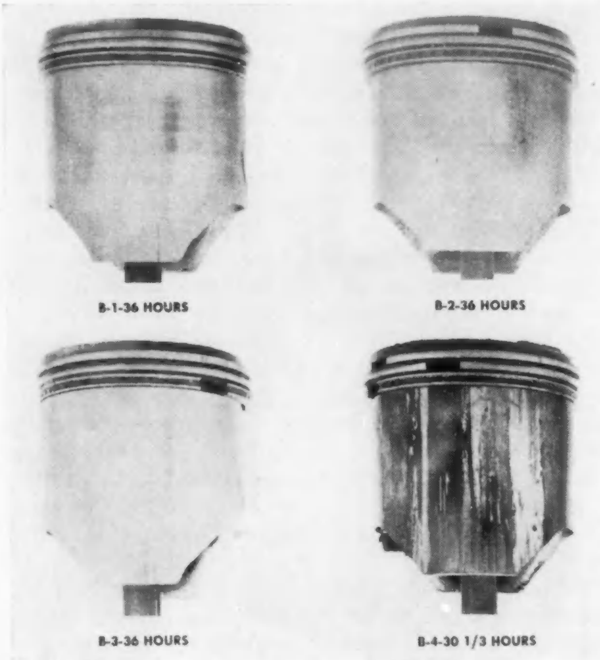
■ Interpretation of Test Results

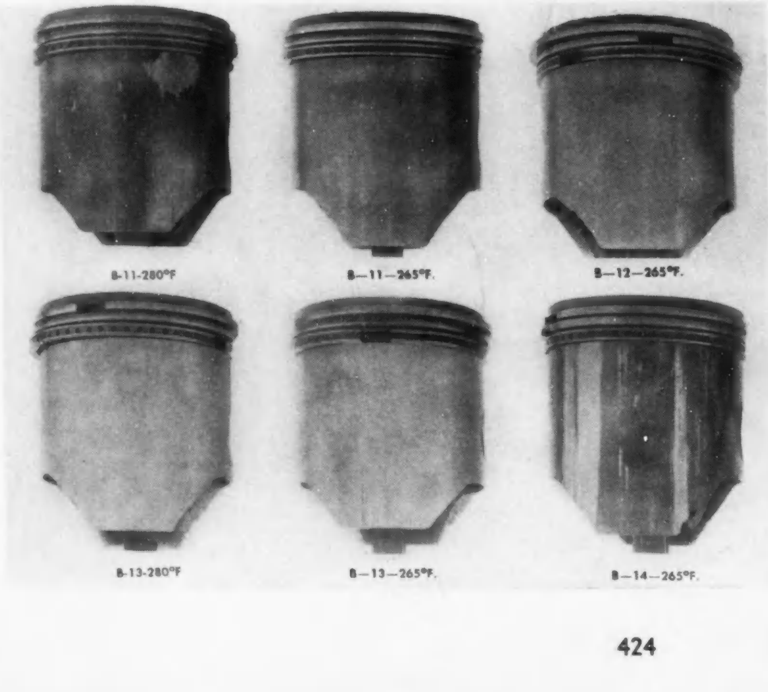
In the cooperative work now under way in Subcommittee B, there are four fundamental questions that must be answered.

1. Is the test procedure sufficiently reproducible so that different laboratories can obtain comparable results?

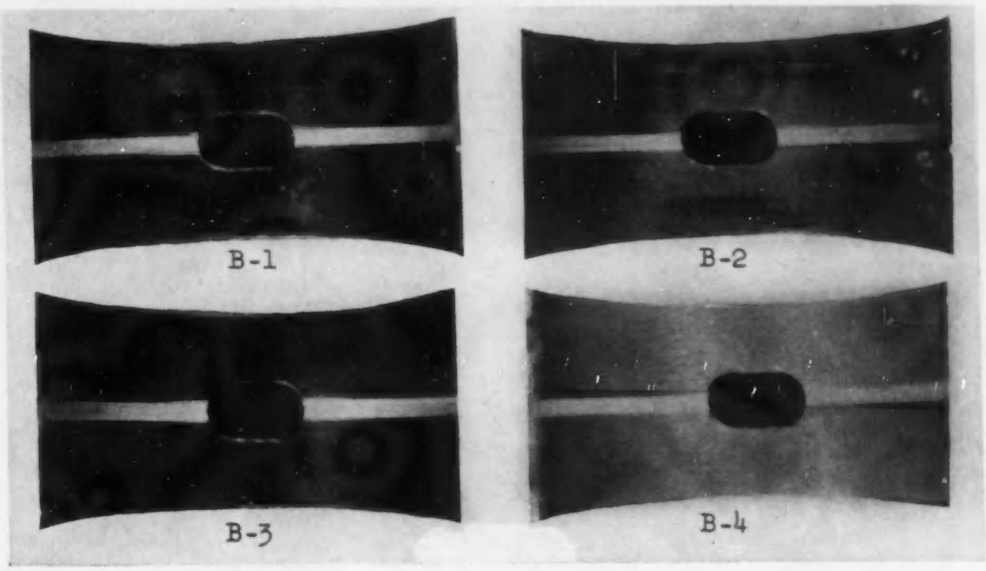
2. Do the test conditions make it possible to show differences between different types of oil?

3. Are the differences that are shown related to performance in service? That is, are the conditions of the test procedure such that the test data may be used to predict performance in service, at least in respect to certain properties of the oils?

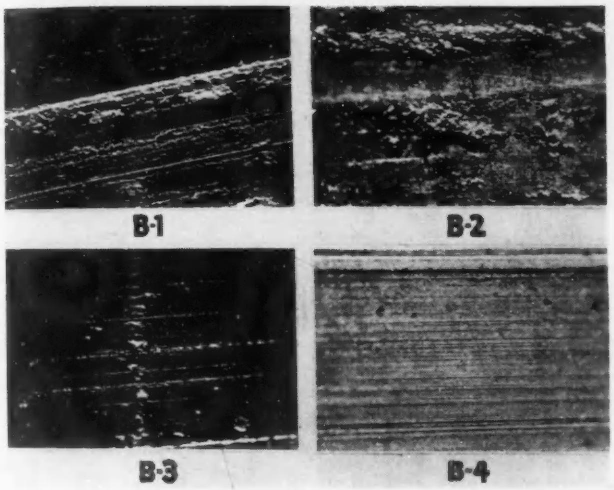




■ Fig. 19 – Pistons after running a given number of hours with B-1, B-2, B-3, and B-4 oils (top) and pistons after running at various temperatures with B-11, B-12, B-13, and B-14 oils



■ Fig. 20 – Bearings from 36-hr tests on B-1, B-2, B-3, and B-4 oils made by Laboratory No. 3



■ Fig. 21 – Enlarged areas of bearing surfaces from 36-hr tests on B-1, B-2, B-3, and B-4 oils made by Laboratory No. 3 – 50X

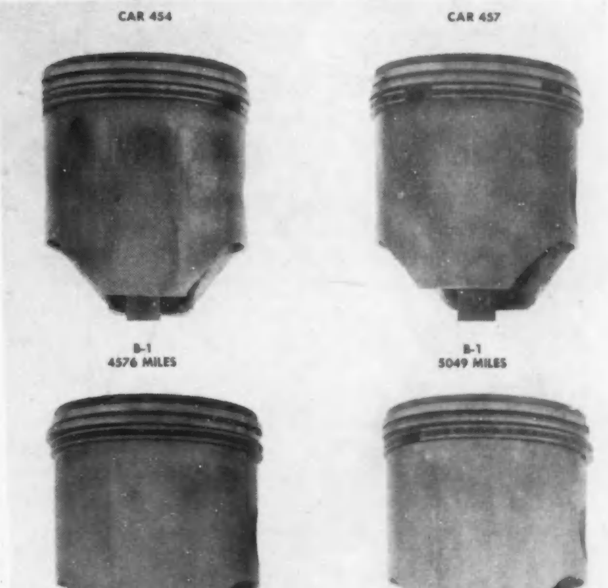
	Dynamometer	Road
Miles	4000	4770
Neutralization No.	4.38	0.83
Naphtha Insoluble	0.94	2.11
Chloroform Soluble	0.18	0.38
Conradson Carbon	2.61	2.65
Ash	0.50	0.65

■ Fig. 22 – Used oil analyses on same oil from dynamometer and road tests (Table 8 from "Crankcase Oils for Heavy-Duty Service," by H. R. Wolf[5])

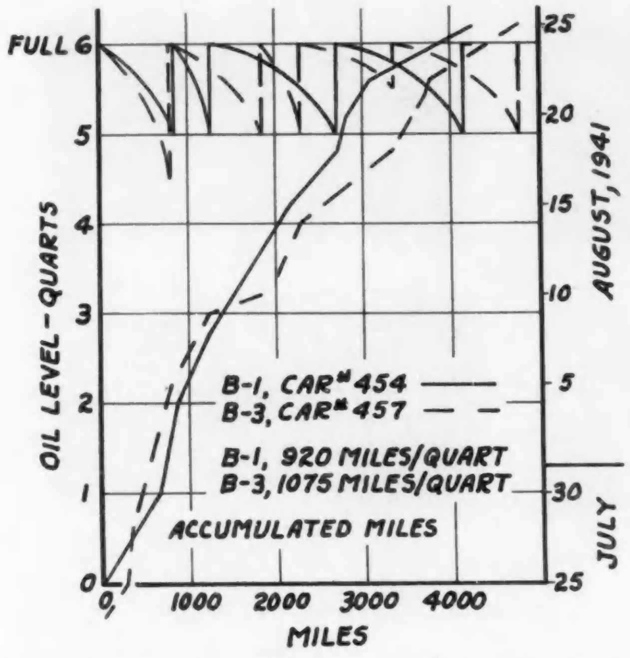
4. What other additional test procedures are necessary before general predictions can be made with regard to the performance of the oils under such widely different operating conditions as city driving with gasoline engines, heavy-duty service with gasoline engines, heavy-duty service with diesel engines, of different types, and so on?

The answer to the first of these questions with regard to the reproducibility of the test has already been given. We believe that it has been shown that the test procedure is reproducible, within and between laboratories.

In regard to Question 2 as to the test procedure showing differences between different types of oils, the answer is given in the data from the different cooperating laboratories and the photographs (Figs. 18 and 19). B-1 and B-3 are resistant to oxidation and give clean engines. B-2 gives a clean engine, but the oil oxidizes badly and is corrosive to copper-lead bearings. B-4 gives a very dirty and badly varnished engine, and the oil oxidizes badly, but the acids that are developed are not appreciably cor-



■ Fig. 23 – Pistons from four road tests made by Laboratory No. 2 using oils B-1 and B-3 in two passenger cars

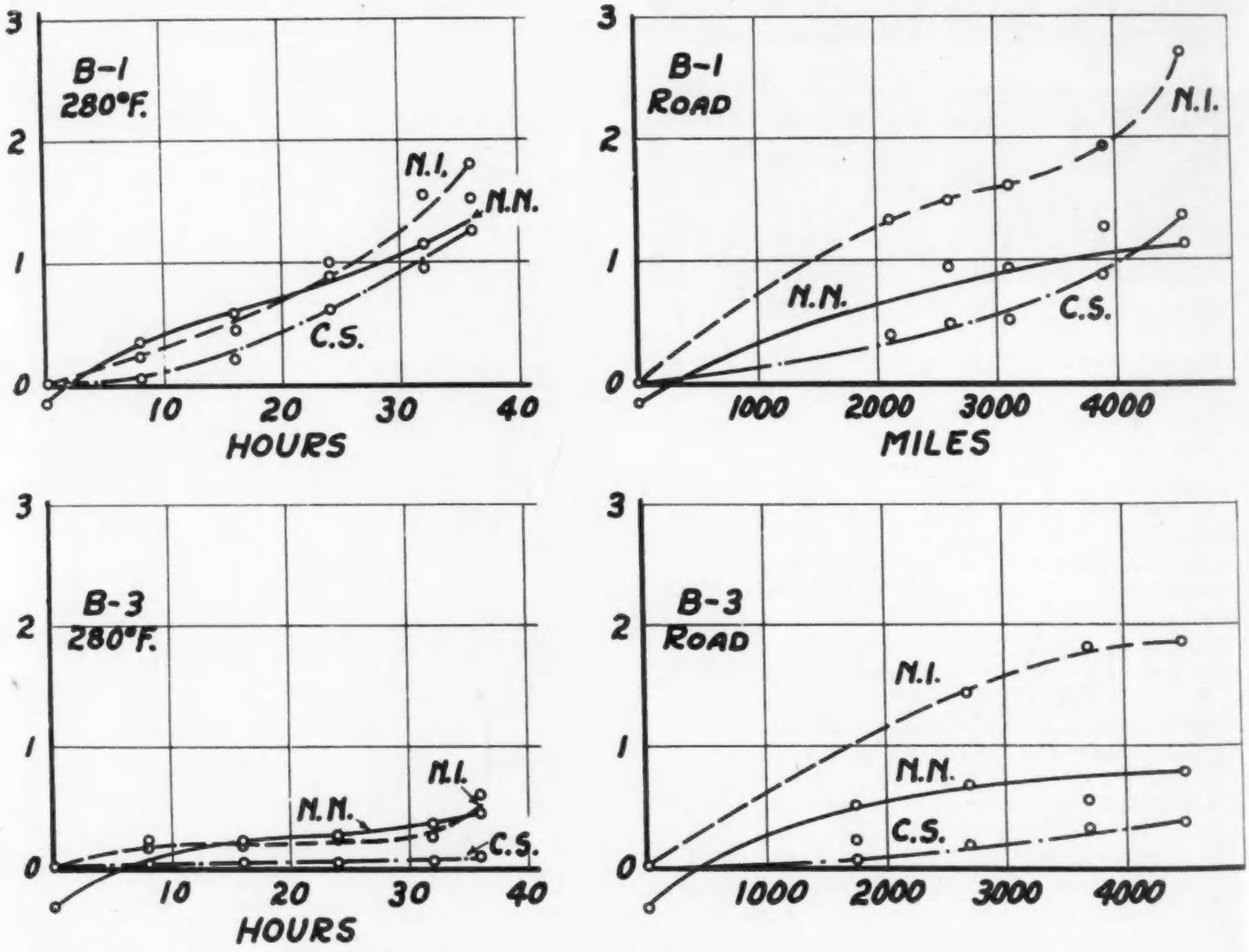


■ Fig. 24 – Operating conditions for one run each on B-1 and B-3 oils of the road test

rosive to copper-lead bearings under the service conditions (see Figs. 20 and 21).

In answer to Question 3 in regard to the relation of the 36-hr test to performance in service, there are considerable data. First, however, it must be recognized that, although oxidation and decomposition of oil are factors of time times temperature, yet the results obtained in a short time at a high temperature will not be exactly the same as those obtained in a longer time at a lower temperature. We are not at all surprised to find that the results are quite different if we take two eggs, both as nearly identical as possible, and put one under a hen for three weeks at approximately 100 F and the other in water for three minutes at 212 F.

In their paper on engine deposits, given at the World Automotive Engineering Congress of the SAE in New York in 1939, Stewart and Story point out that "coffee-grounds" in used engine oil are formed by a series of reactions which involve the oxidation of the oil to form oil-soluble products, the precipitation of these oil-soluble products by the addition of fresh oil and, finally, the conversion of these precipitated products into "coffee-grounds," as the result of their being exposed for some time to relatively moderate temperatures.



■ Fig. 25 – Relation between the used-oil analyses from the 36-hr tests made at 280 F, and one each of the road tests – B-1 and B-3 oils

This same subject is discussed by H. R. Wolf[5] in the April, 1941, SAE Transactions, and Fig. 22 is reproduced from this paper. Mr. Wolf states:

> "On the dynamometer, the neutralization number is generally higher and the naphtha insoluble and chloroform soluble are generally lower than obtained on the same oil on the road when operated for approximately 4000 miles without crankcase oil changes."

However, in comparing data from the 36-hr test with performance on the road in passenger-car service, it will be noted that, if an oil gives high neutralization number results in the 36-hr test, it will give high results on the road and, if the chloroform soluble is either a large or small percentage of the naphtha insoluble in the dynamometer test, it will be a correspondingly large or small percentage of the naphtha insoluble in the road test.

Fig. 23 shows the pistons from four road tests in which oils B-1 and B-3 were used in two different tests each in

[5] See SAE Transactions, April, 1941, pp. 128-137: "Crankcase Oils for Heavy-Duty Service," by H. R. Wolf.

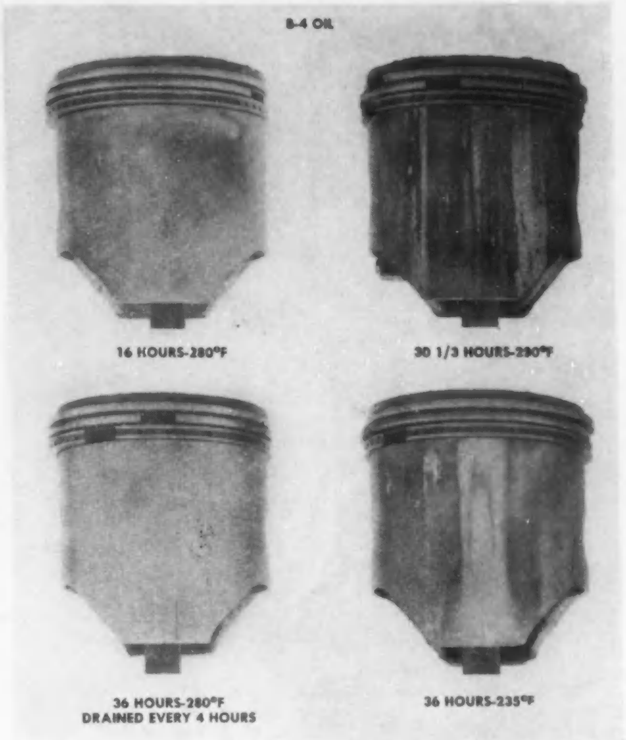
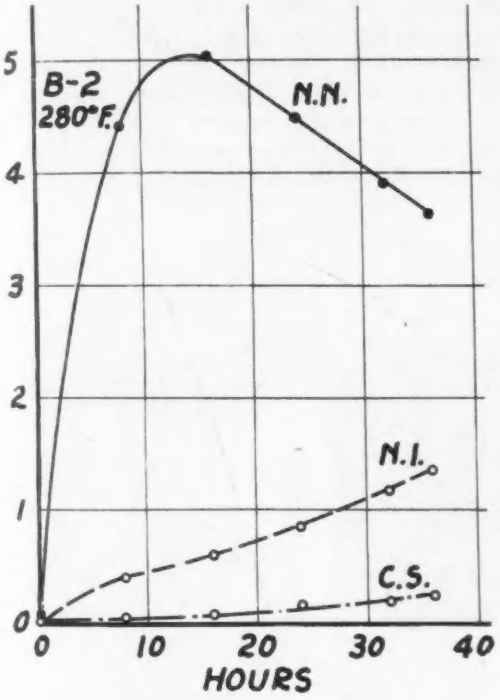
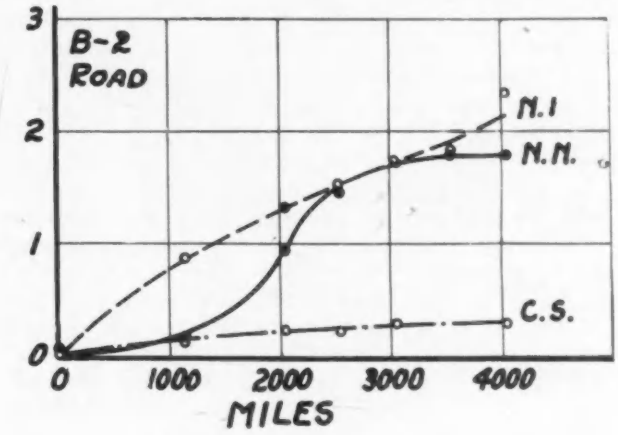
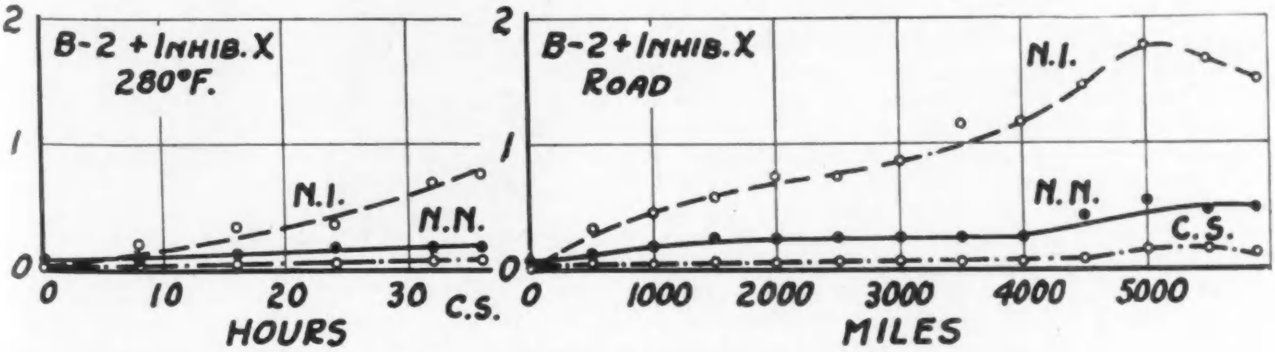


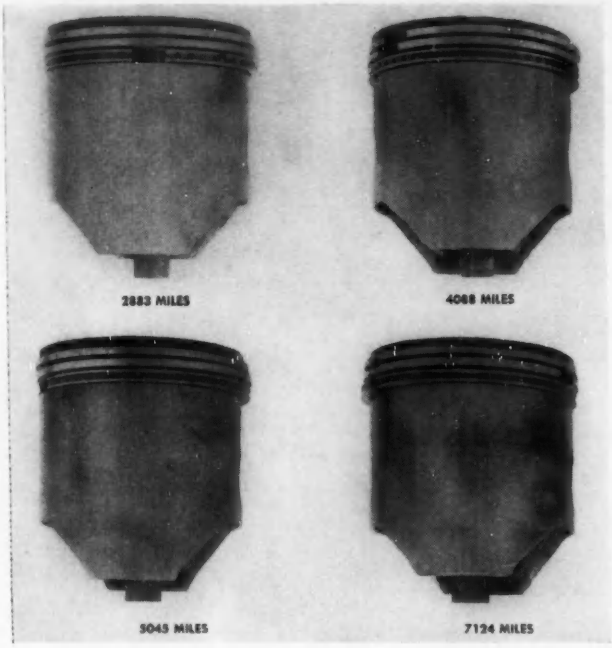
Fig. 27 – Pistons from three different engine tests on B-4 oil







■ Fig. 26 – Relation between the used-oil analyses of B-2 oil from the 36-hr tests at 280 F and same oil operated on the road (upper set of curves); effects of addition of inhibitor to B-2 – 36-hr tests at 280 F and road test



■ Fig. 28 – Appearance of piston at four different mileages in road test on B-4 oil

at 30 1/3 hr at 280 F, and the much better piston which completed 36 hr at an oil temperature of 235 F. Since the pistons in the road test at the end of the 7124 miles are intermediate between the fairly good piston and the bad piston shown in Fig. 27, it is apparent that this road schedule was intermediate between the 36-hr test conditions at 235 F and 280 F. Since oxidation of oil is a factor of time and temperature, it is evident that the average temperature in the road test was not excessive during most of the road test of 7124 miles (which would be 178 hr if the average car speed had been 40 mph).

Fig. 29 shows the relation between the used oil analyses in the dynamometer tests and the used oil analyses from samples taken during the road test on oil B-4. In this Fig. 29, curves are also given showing the improvement in oil analyses resulting from frequent oil changes (every 4 hr), and also the improvement resulting from adding an inhibitor to oil B-4. Fig. 30 shows the corresponding appearances of the pistons and the valve cover plates from two of these tests, one of B-4 oil at 280 F which stuck up at 30 1/3 hr, and the other of B-4 oil plus an inhibitor which gave a clean engine, free from varnish or sludge, when tested for the full period of 36 hr at 280 F.

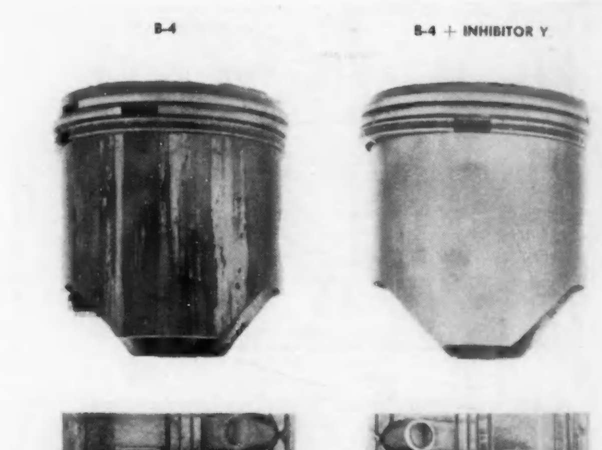
Attention is called to the fact that the neutralization number in the test at 280 F on oil B-4 plus inhibitor Y (Fig. 29), starts at a value of 1.00 and then drops to 0.95 and remains at 0.95 throughout the test. In the test at 280 F on oil B-2 plus inhibitor X (Fig. 26) the neutraliza-

two passenger cars operating on the road. In each of these tests, the oil mileages were from about 4500 to 5000 miles without changing oil. Fig. 24 shows the operating conditions for one run each on B-1 and B-3 of the road tests. Fig. 25 shows the relation between the used-oil analyses from the 36-hr tests made at 280 F and one each of the road tests on oils B-1 and B-3.

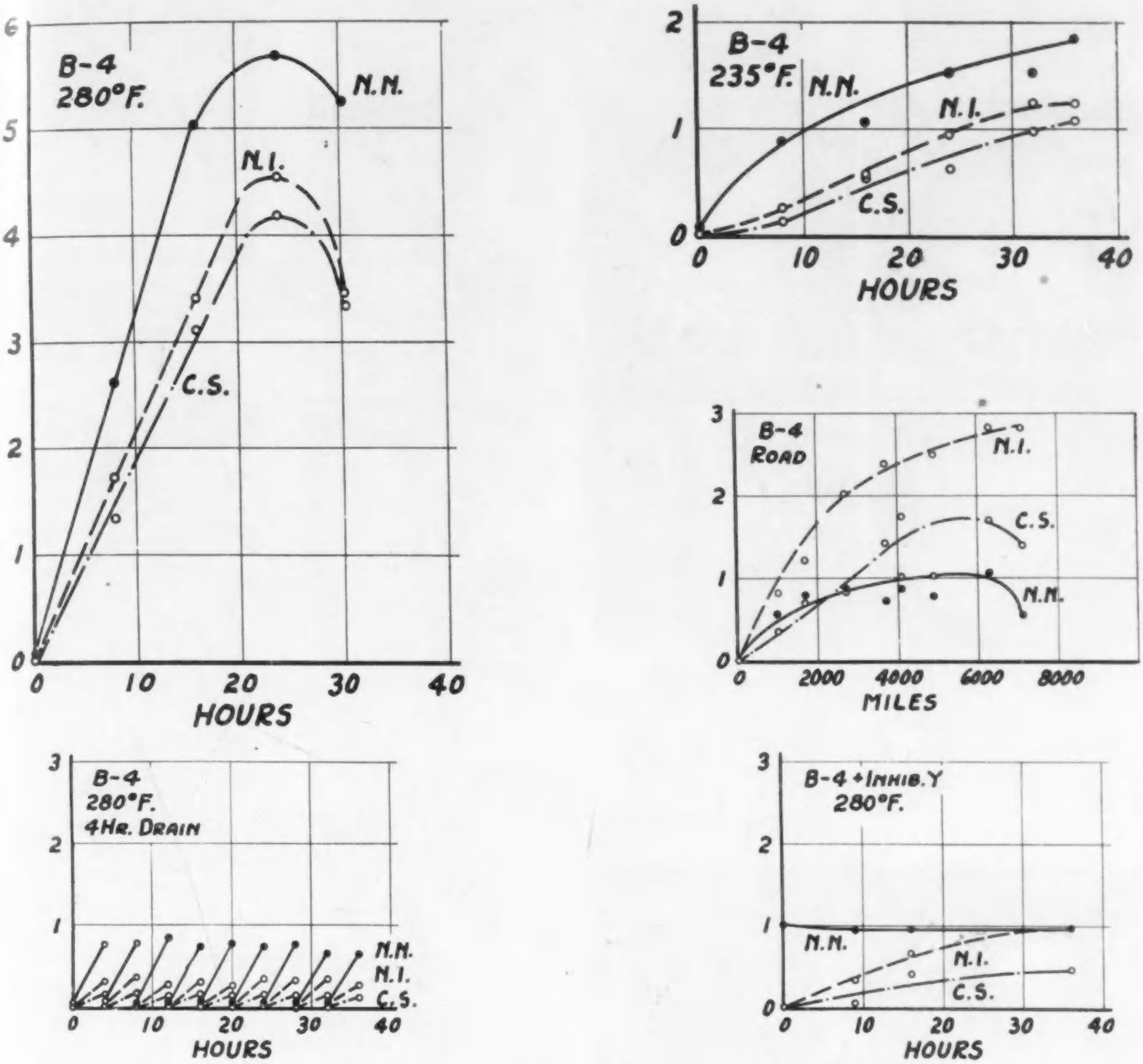
Fig. 26 (upper set of curves) shows the relation between the used-oil analyses from the 36-hr tests at 280 F of B-2 and this same oil, B-2, operated on the road under conditions similar to those shown in Fig. 24. Fig. 26 (lower set of curves) shows the effects of addition of an inhibitor to B-2 and the used-oil analyses of the inhibited oil from both the 36-hr test at 280 F and from the road test, also under conditions similar to Fig. 24.

In Fig. 27 the appearance of the pistons from three different engine tests on oil B-4 is shown. The top-left picture shows the appearance of the piston after 16 hr at 280 F oil. The top-right picture shows the same piston at the end of 30 1/3 hr at 280 F when the engine stuck up due to the varnish on the piston skirts cementing the pistons to the cylinder walls. The lower-left picture shows the very excellent appearance of the piston after another test of 36 hr with 280 F oil temperature, but with the oil drained and replaced every 4 hr. (Nine crankcase fills of oil, each used 4 hr, were required for this 36-hr test.) The lower-right picture shows the appearance of the pistons after a 36-hr test on the same B-4 oil with the test procedure standard in all respects except that the oil temperature was held at 235 F instead of 280 F.

Fig. 28 shows the appearance of the piston at four different mileages in a road test on B-4 oil under operating conditions similar to those shown in Fig. 24. It will be noted that the piston, after 7124 miles without draining the oil, is intermediate in appearance between the badly varnished piston, in Fig. 27, which stuck up due to varnish



■ Fig. 30 – Comparison of pistons and valve cover plates between test of B-4 oil at 280 F for 30 1/3 hr, and test of B-4 oil plus inhibitor Y for 36 hr at 280 F



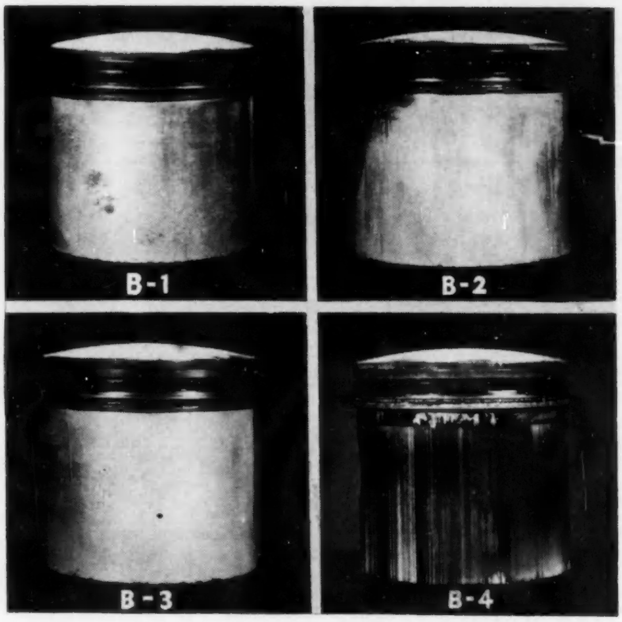
■ Fig. 29 – Relation between used oil analyses in the dynamometer tests and used oil analyses from samples taken during road test – B-4 oil

tion number starts at zero and remains very low throughout the test. The differences in the neutralization numbers of the two oils in these tests are due to the different character of inhibitors X and Y. Oil B-2 is a highly refined high VI oil, and the inhibitor X is strong in inhibitor properties but weak in detergent properties. Oil B-4 is a moderately refined low VI oil, and the inhibitor Y is strong in both inhibitor and detergent properties. Neutralization number, as is well known[6], is a composite measure of many different things, including the amounts of certain inhibitors, and the fact that an oil may have a large neutralization number does not mean that the oil contains any free acid; and a neutralization number, even if it is due to free acid, is not of necessity any indication as to whether or not the oil is corrosive to copper-lead bearings.

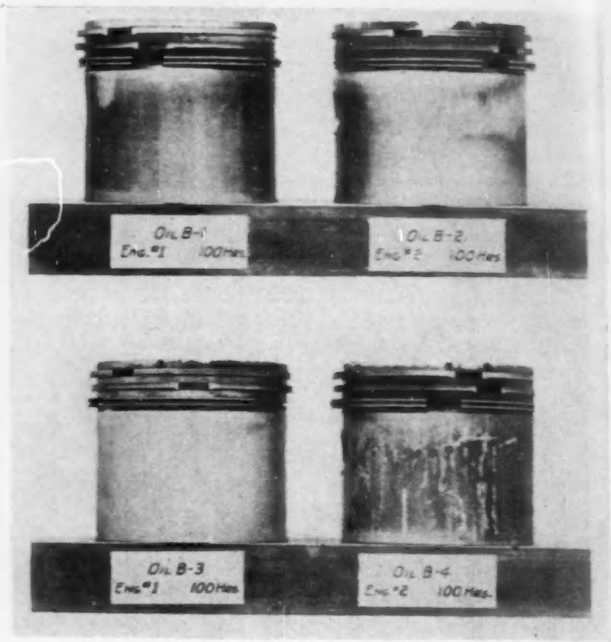
The two sets of curves at the bottom of Fig. 29 and the photographs corresponding to these two tests (the lower left piston at the bottom of Fig. 27, and the right-hand set of pictures, Fig. 30), also show that, while very frequent draining of oil and replacing it with fresh oil is one way of overcoming the troubles due to oil oxidation, there is another way which many people believe to be very good, and that is by the use of an oil which is able to withstand the operating conditions without oxidation. The production of such an oil may involve selection of the crude, methods of refining, and addition agents.

In regard to Question 4, as to what other additional test procedures may be necessary before general predictions can

[6] See p. 13, ASTM Bulletin, December, 1941: "The Examination of Used Engine Crankcase Oil," by L. L. Davis.

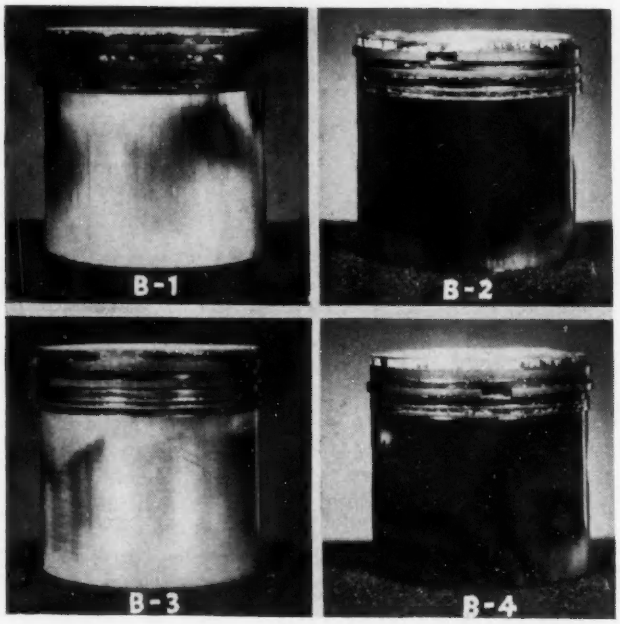


Fuel – 1.9 cc Lead
Laboratory 3



Fuel – O cc Lead, 0.045% S
Laboratory 14

100-Hr Lauson Engine
1600 rpm, 170 F Water, 280 F oil



Laboratory 3

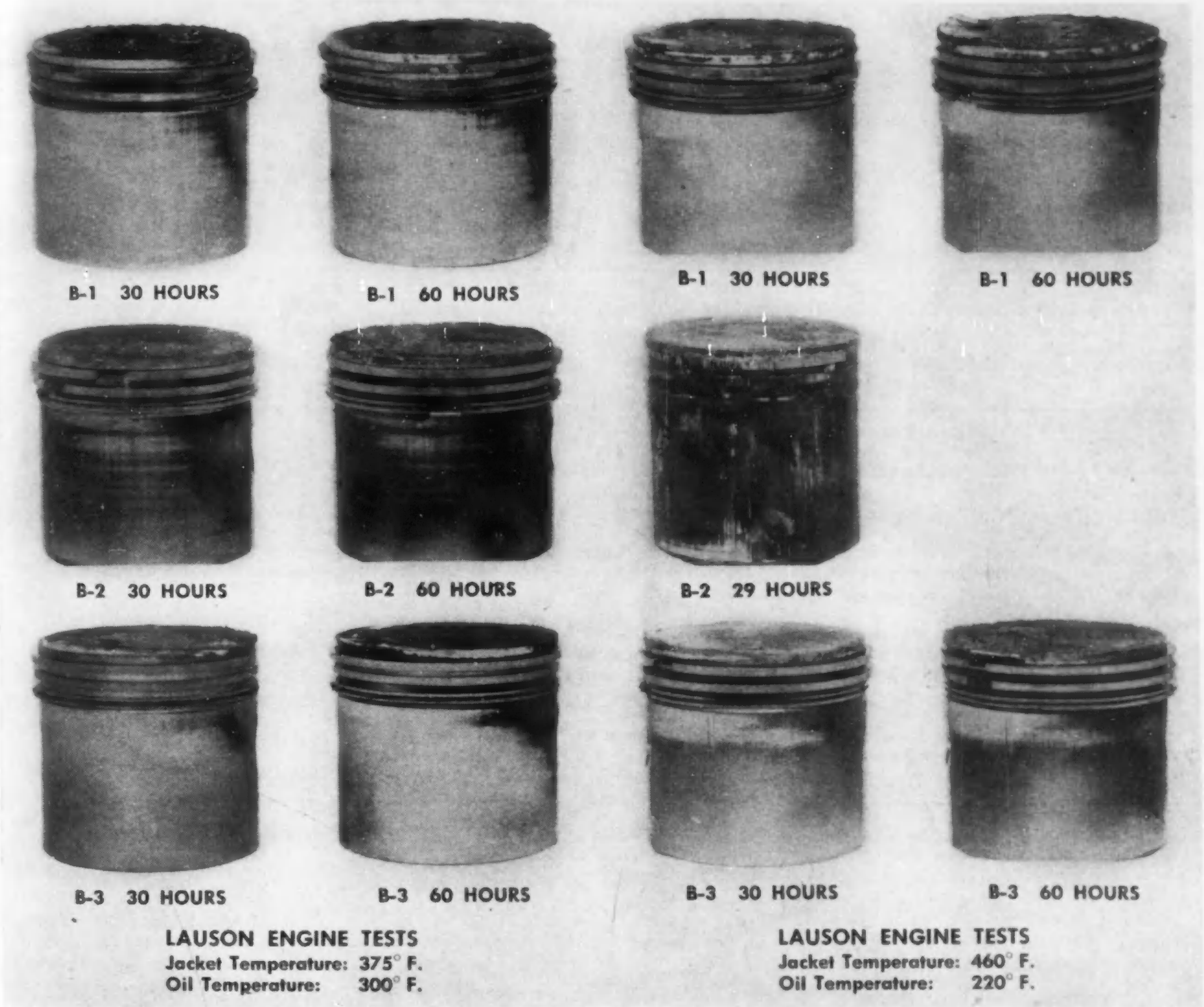
Lauson Ring-Sticking Procedure

New cylinder block and new piston. Break-in, 22 hr, according to a schedule with speed ranging from 1200 rpm to 1800 rpm, and jacket temperature ranging from 220 F to 400 F. Engine is then cleaned and ring-sticking test is made under following conditions:

Engine Speed	1800 rpm
Load	1 kw
Jacket Temperature	400 F ± 5 F
Oil Sump Temperature	225 F ± 8 F
Air-Fuel Ratio	12.8:1
Fuel (regular gasoline)	1.9 cc tel/gal

Results of tests on B-1 to B-4, inclusive, shown at left. B-3 and B-1 passed the test in this order with all rings free and the skirt quite clean. Oils B-2 and B-4 failed in this order; B-2 has the top ring cold-stuck with the second ring and the oil ring hot-stuck about 50/60 deg. B-4 has the top and the oil rings hot-stuck with the second ring extremely sluggish.

■ **Fig. 31 – Pistons from tests on B-1, B-2, B-3, and B-4 oils made on Lauson engine**



■ Fig. 32 – Pistons from additional Lauson engine tests – B-1, B-2, and B-3, oils – Laboratory No. 16

be made in regard to the performance of an oil in engines of different kinds and operated under different conditions of service, there are also considerable data. As previously pointed out in this paper, the 36-hr test is a measure of oxidation resistance without very much importance attached to detergency, and the Caterpillar test is a measure of detergency without much importance attached to oxidation resistance. Whether or not an oil needs oxidation resistance or detergency, or both, depends upon the design of the engine and the operating conditions. In general[5], an oil for a gasoline engine is more likely to need oxidation resistance without much importance attached to detergency, and an oil for a diesel engine is more likely to need detergency with less importance attached to oxidation resistance, but these are very general statements and they are not universally true; under many operating conditions, both properties in varying degrees are desired in oils for both kinds of engines. On account of the varying effects of these factors, it is necessary that both oxidation resistance and detergency be evaluated before predictions can be made in regard to performance in service. As illustrations of the manner in which differences in operating conditions place different emphasis on these two factors, Figs. 31 to 33, inclusive, should be studied.

Fig. 31, top-left group of pictures, shows how the reference oils B-1 to B-4, inclusive, perform in a Lauson engine under test conditions approximating those of the 36-Hr Oxidation Test, and the top-right group of pictures shows how similar results are obtained under similar conditions in another laboratory. These pictures, together with many other pictures and data shown in this paper, illustrate the point that, under conditions of moderate piston temperature and high crankcase oil temperature, oxidation resistance is very important so that corrosion or varnish, or both, may be avoided, but detergency is of minor importance.

In the lower group of pictures in Fig. 31, the test conditions are reversed; the piston temperature is increased and the crankcase oil temperature is decreased. Under these test conditions, an oil with good detergent properties, such as B-1 or B-3, will give good results but, if detergent properties are lacking, an oil like B-2 will cause ring sticking and bad carbon troubles, and an oil like B-4 may cause troubles due to varnish and sludge and may also cause ring sticking.

Fig. 32 shows the results obtained by still another lab-

(Pistons shown in Figs. 31 and 32)

Lauson Engine Test Conditions				Piston Appearance			
Jacket Temp. °F.	Crankcase Oil Temp. °F.	Time for Test (Hrs.)	Oil Drain Periods (Hrs.)	B-1 Medium VI oil Contains both oxidation inhibitor and detergent.	B-2 High VI oil. Highly refined. Does not contain oxidation inhibitor or detergent.	B-3 High VI oil. Contains both oxidation inhibitor and detergent.	B-4 Low VI Medium refined. Does not contain either oxidation inhibitor or detergent.
170	280	100	100	Fairly good	good	good	bad*
375	300	30	30	good	bad*	good	--
375	300	60	30	good	bad*	good	--
400	225	25	25	good	bad**	good	bad**
460	220	30	30	good	very bad***	fairly good	--
460	220	60	30	good	--	fair	--

* Piston badly varnished but no rings stuck.
** Piston badly varnished, some ring sticking.
*** Piston badly varnished, rings stuck very badly.

NOTE: In the tabulation, in describing the properties of the oils B-1 to B-4, inc., the nature of the base oil is not described, except as to VI. However, in general, low VI oils contain some natural inhibitors and they have a certain amount of natural detergent properties. High VI oils, especially when they are highly refined, tend to be low in both natural oxidation inhibitor and detergent properties. In this tabulation and in the pictures (Figs. 31 and 32) the points illustrated are -

1. Low piston temperature, (resulting from low jacket temperature under the operating conditions in the Lauson engine test procedure) combined with high crankcase oil temperature, measures oxidation resistance of the oil, without showing much effect of detergency.
2. High piston temperature, combined with low crankcase oil temperature, measures detergency, without showing much effect of oxidation resistance.
3. High piston and crankcase oil temperatures, both at the same time, measure both oxidation resistance and detergency, but these effects cannot be separated by this test procedure.

■ **Fig. 33 – Important variables with regard to piston and crankcase oil temperatures for the tests illustrated in Figs. 31 and 32, and the effects of these variables**

oratory using oils B-1 to B-3, inclusive, in the Lauson engine under still other combinations of test conditions, changing both piston temperature and crankcase oil temperature. It will be noted that, in these tests also, there is the same general pattern as to the effects of these variables. In Fig. 33 the important variables in regard to piston and crankcase oil temperature illustrated in these tests are tabulated, and the effects of these variables are shown.

The photographs shown in Fig. 34 are illustrations of the results obtained in heavy-duty diesel service by use of oils of different types in one make of diesel which is intermediate in its requirements in regard to both oxidation resistance and detergency. These pistons are from 500-hr tests, made without any oil drains, in accord with the regular Detroit Diesel Division Test Procedure. Briefly, the test conditions are:

Length of test, hr – 500
Rpm – 2000
(Equivalent mph) – 60
Load per cylinder (bhp) – 27½
Jacket temperature, F – 180
Oil temperature, F – 230
Approximate oil consumption (lb per hr per cyl) – 0.06
(Equivalent mpq, 6-cyl, 165 bhp, 60 mph, 2000 rpm) – 300
(Equivalent miles, without oil drain) – 30,000
Bearings – Copper-lead
Filters – no filter up to 144 hr (8640 miles). Filter installed at 144 hr, and changed during test if necessary.

It will be noted that the oxidation resistance and the detergency of both B-1 and B-3 are sufficient to prevent bearing corrosion and ring sticking in these tests. The test on an oil similar to B-2, but with some oxidation inhibitor and some detergent, illustrates the point that an oil like B-2 without either oxidation inhibitor or detergent would probably fail badly in both bearing corrosion and ring sticking. The test on an oil similar to B-4, except that it contains both oxidation inhibitor and detergent, illustrates the point that "piston scuffing" is a function of a number of factors, including the nature of the base oil, since similar "piston scuffing" would be obtained, regardless of the presence or absence of oxidation inhibitor or detergent, with a base oil of this character. This time of 17½ hr was too short for the effects of the properties of oxidation resistance and detergency to become apparent. However, it is believed that, when the problem of "piston scuffing" is better understood, it may be possible to correct this difficulty by the use of suitable addition agents.

These tests shown in Fig. 34 further illustrate the point that, when the conditions of service require different degrees of different properties, it is impossible to predict the performance of an oil in service when all of these properties have not been evaluated. Although it is possible to predict the performance of an oil in the Detroit Diesel Division Test when the various factors of oxidation resistance, detergency, resistance to "piston scuffing," and so on, are known, without making an actual test on this engine, such a prediction cannot be made as the result of a test on a single property, such as oxidation resistance or

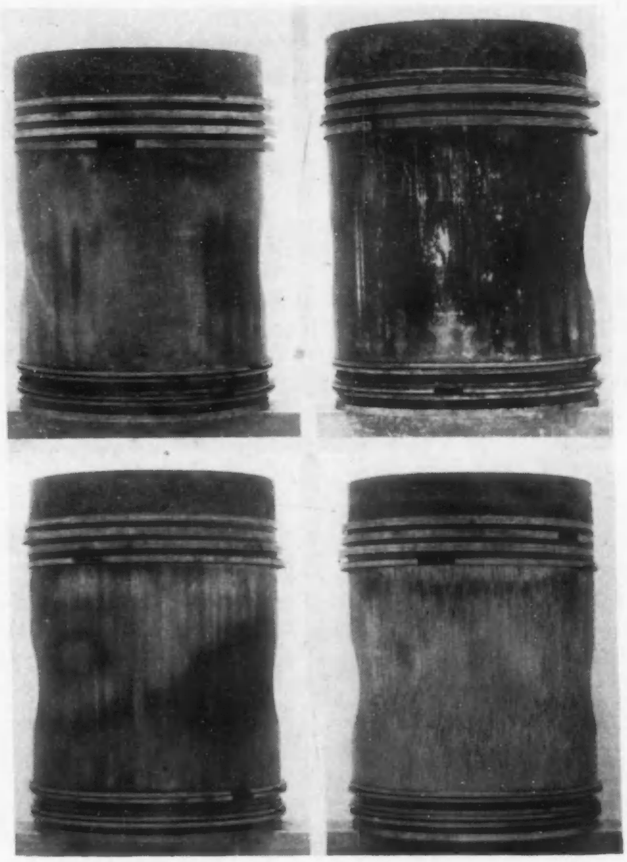
detergency alone. In a similar manner, an oil may fail badly under the conditions of the 500-Hr Diesel Test, and yet the oil may be very satisfactory in gasoline engine service or in a diesel operated under different conditions. Attention is further called to the fact that, when these heavy-duty test conditions are changed to sufficiently moderate conditions, then most or all of these oils will give excellent results.

The obvious conclusions from these data are that, when the difficulty is due to high crankcase oil temperature without high piston temperature (a condition more likely to be encountered in gasoline engines than in diesel engines), then oxidation resistance is very important and detergency is of less importance. When the difficulty is due to high piston temperature without high crankcase oil temperature (a condition more likely to be encountered in diesels than in gasoline engines), then detergency is very important and oxidation resistance is of less importance. With both high piston temperature and high oil temperature (a condition more likely to be encountered in both diesels and gasoline engines in very "heavy-duty" service), then both oxidation resistance and detergency are important.

While a single set of test conditions could be devised so that both oxidation resistance and detergency are necessary in an oil, the use of such a test procedure would eliminate all oils which do not have both high detergency and high oxidation resistance. For the rapid evaluation of addition agents, such a combined test procedure might be desirable. However, much more information is probably obtained from the two test procedures now available, one that measures oxidation resistance and one that measures detergency. The Caterpillar test is now universally accepted as the standard detergency test, and it is to be hoped that the 36-Hr Oxidation Test will provide the other standard procedure.

Since the volume of lubricating oil used in city driving is so many times greater than that used in diesels, and since we do not have any recognized tests to evaluate oils in regard to their performance under light city driving conditions, when the temperatures of the crankcases, combustion chambers, pistons, jackets, and so on, may be low, it may be that the next problem to be studied will be the evaluation of oil under conditions of light city driving.

It would seem apparent that a combination of heavy-duty oxidation resistance and detergency tests alone does not tell everything about an oil. Under certain conditions there are other problems, such as exhaust-valve sticking, piston scuffing, formation of carbon in the combustion chamber, and so on. In general, these are minor problems, and it is always possible to study them in any particular engine operated under a particular set of operating conditions by using the oil in the engine. In addition, experience has shown that certain general types of oils are supe-



B-1, 500 hr, no bearing corrosion, no ring sticking.

An oil similar to B-2, 500 hr, except that some oxidation inhibitor and some detergent have been added. The additions are on the low side. There was a very small amount of bearing corrosion and one ring was stuck. More inhibitor and more detergent would produce better results in both respects.

B-3, 500 hr, no bearing corrosion, no ring sticking.

An oil similar to B-4, 17½ hr, except that this oil contained both oxidation inhibitor and detergent. Test stopped due to bad "piston scuffing." Similar results would be obtained, regardless of presence or absence of oxidation inhibitor and detergent, since "piston scuffing" is due to nature of the base oil.

■ **Fig. 34 – Heavy-duty diesel pistons from 500-hr tests in accord with the Detroit Diesel Division Test Procedure – oils of different types tested in one make of diesel**

rior in these different respects, although an oil that is good in one of these properties may be poor in another.

These facts were discussed by A. E. Smith and J. P. Stewart at the SAE National Fuels and Lubricants Meeting at Tulsa, Okla., Oct. 23, 1941. They pointed out that, although low VI oils, in general, will give less "carbon" trouble, high VI oils are the kind that are most likely to get us out of "piston-scuffing" trouble. Since "piston scuffing" is a difficulty that is affected by a number of factors, such as the chemical and physical properties of the piston and cylinder combination, the bmep, the speed, the temperature, the nature of the "break-in," and so on, it may be that "piston scuffing" will yield to the use of the right kind of addition agents after we know more about it. However, since these are somewhat minor problems at the present time, they should not be allowed to interfere with the much more important problems in regard to the establishment of a common yardstick for the determination of oxidation resistance on the one hand and detergency on the other.

Of necessity, the laboratory and field data herewith presented have been brief. Much data have been omitted, not only for lack of time and space, but also because the bulk of the field data has been collected under trade or brand names which, while making it valuable from an informative point of view, prohibits its use in this type of a discussion. The statement can be made, however, that these data indicate the same findings, in the same order and relative magnitude, as the data already discussed.

We should like to again point out and emphasize the fact that the basis upon which this work was started and will continue, is: (1) that of developing a laboratory test procedure for corrosion and oxidation, which would rate oils as they are found to be rated in service; and (2) that the Caterpillar series of tests adequately defines a motor oil for detergent and dispersive properties.

It should be emphasized again that the purpose of the 36-Hr Oxidation Test Procedure plus the Caterpillar Test Series is for the definition and specification of heavy-duty motor oils for gasoline and diesel engines. For milder service, other oils may be not only entirely satisfactory, but may be even more desirable.

In conclusion, we believe that:

1. By means of carefully controlled operating conditions, production multicylinder engines can be used, in conjunction with a series of standard test oils, to give a test pattern for these oils which are reproducible themselves, and which will coincide with the results obtained on these oils in service in so far as corrosion and oxidation resistance characteristics are concerned;

2. The description and specification of heavy-duty motor oils of known and unknown performance for oxidation and corrosion characteristics are, for the moment at least, most easily achieved by describing this oil in terms of a reference oil test pattern;

3. The 36-Hr Oxidation Test Procedure (see Table 5, Appendix) is a satisfactory method for the evaluation of oils for heavy-duty service with respect to their oxidation and corrosion characteristics only; and an additional test procedure, such as the Caterpillar Test series, is required to measure certain other properties, such as resistance to ring sticking, resistance to port clogging, resistance to piston scuffing and cylinder scratching, and so on;

4. Therefore, by means of the 36-Hr Test Procedure and the Caterpillar Test Series, a heavy-duty motor oil may be evaluated for most of the important characteristics; however, valve sticking, winter sludge, and other undesirable characteristics are not necessarily evaluated by these tests and, if these factors become important, a separate study will have to be made for their evaluation;

5. From the point of view of the engine manufacturer, not only can unknown oils be evaluated by means of the known reference test oil pattern, but so can engines of present and future designs; the severity of the conditions which the engine imposes on the oil may also be evaluated for the declared drain time;

6. To make these test procedures practical, an Engine Test Committee must continuously, and continually, follow the progress in both engine and oil design so that, once such a test procedure and description have been decided upon and accepted, changes such as must necessarily be demanded by progress shall be made only with the consent and approval of all those affected by such changes.

■ Test Data Records

In Tables 1 to 4 inclusive, the data are given from oils B-11 to B-14, inclusive. The data for oils B-1 to B-4, inclusive, were given in the report[4] presented at the API meeting, San Francisco, Nov. 3-7, 1941, and consequently these data are not repeated in this report.

■ Acknowledgment

The work covered in this paper was done by a number of cooperating laboratories working under the auspices of the Lubricants Division of the SAE Standards Committee. E. W. Upham of Chrysler Corp., is chairman. The cooperating laboratories are listed below. The order of listing does not bear any relation to the laboratory numbers by which the laboratories are identified in the paper:

Atlantic Refining Co.
Carbide and Carbon Chemicals Corp.
Caterpillar Tractor Co.
Continental Oil Co.
E. I. du Pont de Nemours and Co., Inc.
General Motors Corp.
Gulf Refining Co.
Kendall Refining Co.
Lubri-Zol Corp.
The Pure Oil Co.
Quaker State Oil Refining Corp.
Shell Oil Co.
Sinclair Refining Co.
Socony-Vacuum Oil Co.
Standard Oil Co. of Calif.
Standard Oil Development Co.
Standard Oil Co. of Ohio
Sun Oil Co.
The Texas Co.
Tide Water Associated Oil Co.

APPENDIX

Table 1 – Comparative Test Data on B-11 Oil
36-hr Results

Laboratory	Engine	V/100 New	V/100 Used	Con. Carb. New	Con. Carb. Used	% Ash New	% Ash Used	Acid No. New	Acid No. Used	Nap. Insol., mg/10 g New	Nap. Insol., mg/10 g Used	Chl. Sol., mg/10 g New	Chl. Sol., mg/10 g Used	Oil Consumption, mpq	Lead in Fuel, cc	Fuel Consumption, 300 cc, min	Intake Manifold Vacuum, in. hg Start	Intake Manifold Vacuum, in. hg End	Exhaust Back Pressure, in. hg Start	Exhaust Back Pressure, in. hg End
2	1941a	198	589	....	2.05	0.14	0.57	−0.11	0.94	....	354	....	270	....	1.9	....	....	....	....	....
	1941b	198	423	....	2.35	0.14	0.62	−0.11	0.61	....	223	....	136	575	1.9	1.126	....	....	....	....
3	1941	203	465	0.25	2.4	0.15	0.60	0.321	1.74	13.0	202.4	10.6	172.2	240	1.37	1.285	10.6	....	2.6	....
4	1941	196	567	0.23	3.50	0.14	0.79	0.05	0.70	....	228	....	177	540	1.5	1.155	12.4	11.8	1.5	1.0
6	1941	203	387	0.27	1.96	0.1496	0.2654	Alk	0.50	9.0	145	5.0	117	730	1.2	1.1	13.4	13.1	.75	.75
8	1941	193.8	459	0.22	2.42	0.15	0.38	Alk	2.8	12	222	4	188	1515	....	....	....	....	....	....
9	1941	195	392	0.23	1.70	0.14	0.44	−0.025	2.45	0	90	0	73	334	....	1.43	....	....	1.1	....
11	1941	194.2	582.1	0.141	2.193	0.148	0.502	0.025	3.79	6	254	3	217	376	1	1.32	12.2	12.2	1.4	1.4
12[3]	1941	....	....	....	....	....	....	....	....	....	....	....	....	626	1.58	1.41	14.3	....	....	....
13	1941	193.8	508.4	0.23	2.40	0.14	0.566	−0.22	1.69	....	210.7	....	184	529	1.24	1.25	11.7	11.5	1.0	1.0
18	1942	....	....	....	1.71	....	0.37	....	6.78	....	226	....	144	625	2/2.5	1.334	14.2	14.5	....	....

Note: Test a, oil temp. 280 F, Test b, 265 F. These two tests were run to show effect of oil temperature.

Laboratory	Engine	Air Fuel Ratio Start	Air Fuel Ratio End	Spark Advance, deg BTDC	Ring Wear, in. No. 5 Piston Top	Ring Wear, in. No. 5 Piston Middle	Ring Wear, in. No. 5 Piston Bottom	Bearing Loss, g Top	Bearing Loss, g Bottom	Bearing Loss, g Average	Engine Rating, % Varnish	Engine Rating, % Sludge	Engine Rating, % Total	Oil Pressure, psi	Sulfur in Fuel, %
2	1941	14.4	15.0	..	.....	.....	.....	0.27 0.33	0.27 0.28	0.27 0.305	....	....	....	....	0.06
	1941	14	13.87	54	0.004	0.005	0.005	0.24 0.26	0.29 0.31	0.265 0.285	38	46.8	42.4	....	0.06
3	1941	16.6	....	50	0.020	0.018	0.023	0.0841 0.1296	0.0919 0.0915	0.0880 0.1106	79.5	63.5	74	16.8	0.039
4	1941	14.2	14.1	52	0.002	0.001	0.004	0.241 0.240	0.254 0.168	0.2475 0.204	93.5	85	90.5	11/12	0.03
6	1941	14.2	13.9	..	0.002	0.002	0.005	0.3752 0.3730	0.3510 0.3203	0.3631 0.3436	74.8	58.6	66.4	16.3	0.043
8	1941	....	....	..	0.008	0.008	0.014	0.187 0.206	0.225 0.207	0.206 0.2065	43.5	60	50.25	....	....
9	1941	15.0	....	52	0.004	0.006	0.007	0.5865 0.4767	0.5594 0.3992	0.57295 0.43795	93.25	72	84.5	....	....
11	1941	14.2	14.4	..	0.003	0.003	0.005	0.165 0.3247	0.135 0.099	0.150 0.2118	77.8	80.7	79	....	0.0517
12[3]	1941	14.2	14.5	..	.....	.....	.....	0.3361 0.3094	0.3834 0.3418	0.3597 0.3256	89.6	82.6	86.9	....	0.03
13	1941	14.3	....	..	0.006	0.005	0.006	0.14 0.15	0.17 0.20	0.155 0.175	80	84	81.5	12-15	0.035
18	1942	14.8	....	55	0.004	0.004	0.003	0.277 0.290	0.358 0.324	0.318 0.307	87	94	85.5	16.2	0.037

[3]Data are incomplete; remainder will follow.

Table 2 – Comparative Test Data on B-12 Oil
36-hr Results

Laboratory	Engine	V/100 New	V/100 Used	Con. Carb. New	Con. Carb. Used	% Ash New	% Ash Used	Acid No. New	Acid No. Used	Nap. Insol., mg/10 g New	Nap. Insol., mg/10 g Used	Chl. Sol., mg/10 g New	Chl. Sol., mg/10 g Used	Oil Consumption, mpq	Lead in Fuel, cc	Fuel Consumption, 300 cc, min	Intake Manifold Vacuum, in. hg Start	Intake Manifold Vacuum, in. hg End	Exhaust Back Pressure, in. hg Start	Exhaust Back Pressure, in. hg End
2	1941	176	333	....	1.47	0	0.15	0.03	4.12	....	107	....	52	575	1.9	1.144	....	....	....	....
3[5]	1941	175.4	384	0.018	.8	None	0.27	0.104	10.00	3.6	61.3	3.2	30	480	1.37	....	11.8	....	1.1	....
4	1941	175	406	0.04	2.75	Nil	0.53	....	1.9	....	172	....	54	965	1.5	1.31	11.7	11.2	1.8	1.5
6	1941	176	546	0.032	3.37	None	0.6110	0.05	5.7	3.6	174	2.2	76	2834	1.2	1.11	14.0	13.9	.75	.88
11	1941	171.3	452.8	Nil	1.835	0.01	0.083	0.025	14.09	2	119	1	31	455	1	1.381	14.1	14.2	1.4	1.4
9	1941	174	336	0.01	1.90	0.001	0.46	−0.025	6.55	0	130	0	50	690	....	1.455	....	....	0.8	....
12[3]	1941	....	....	....	....	....	....	....	....	....	....	....	....	800	1.58	1.24	15.5	14.2	....	....
13	1941	173.8	286	0.006	1.70	0	0.316	0.004	2.20	....	122.2	....	75.9	1340	1.24	1.276	12.3	10.3	1	1.1
18	1942	....	....	....	1.81	....	0.50	....	3.53	....	60	....	6	980	2/2.5	1.315	15	13.8	....	....

Laboratory	Engine	Air Fuel Ratio Start	Air Fuel Ratio End	Spark Advance, deg BTDC	Ring Wear, in. No. 5 Piston Top	Ring Wear, in. No. 5 Piston Middle	Ring Wear, in. No. 5 Piston Bottom	Bearing Loss, g Top	Bearing Loss, g Bottom	Bearing Loss, g Average	Engine Rating, % Varnish	Engine Rating, % Sludge	Engine Rating, % Total	Oil Pressure, psi	Sulfur in Fuel, %
2	1941	14.9	15.05	47	0.001	0.001	0.003	0.91 0.70	0.50 0.75	0.705 0.725	55.75	60	57.5	....	0.06
3[5]	1941	....	....	50	0.006	0.006	0.015	0.7555 0.4716	0.6866 0.5689	0.7215 0.5203	94.5	94.5	94.5	16.25	0.039
4	1941	14.0	....	52	0.002	0.002	0.002	2.745 1.575	1.883 2.080	2.314 1.827	95	92.5	94.5	13/14	0.03
6	1941	13.4	15.2	..	0.003	0.002	0.003	1.8656 2.0820	1.9586 1.7438	1.9118 1.9129	66	50.6	60	16/17	0.043
11	1941	14.2	14.2	..	0.004	0.004	0.011	1.076 0.625	0.7253 0.7510	0.9007 0.6880	73	57.4	67	....	0.0517
9	1941	14.6	....	52	.....	.....	.....	2.5977 2.5176	2.7748 2.5437	2.68625 2.53065	87.2	78.6	83.6	....	....
12[3]	1941	14.1	13.8	..	0.002	0.003	0.006	0.9508 1.0155	1.0022 0.9766	0.9765 0.9960	63.1	54.7	59.7	....	0.03
13	1941	14.4	16.1	..	0.005	0.006	0.006	1.68 1.00	1.77 1.17	1.725[4] 1.085[4]	84.75	89.25	85.8	12-15	0.035
18	1942	14.5	13.8	55	0.005	0.004	0.003	2.467 2.694	2.431 2.732	2.449 2.713	89.5	88	89	15.7	0.037

[3]Data are incomplete; remainder will follow.
[4]Bearing weight taken at 26.5 hr due to excessive noise and pounding.
[5]Run stopped at 8 hr so that data could be obtained for paper. Acid No. was 10.

Table 3 – Cooperative Test Data on B-13 Oil
36-hr Results

Laboratory	Engine	V/100 New	V/100 Used	Con. Carb. New	Con. Carb. Used	% Ash New	% Ash Used	Acid No. New	Acid No. Used	Nap. Insol., mg/10 g New	Nap. Insol., mg/10 g Used	Chl. Sol., mg/10 g New	Chl. Sol., mg/10 g Used	Oil Consumption, mpq	Lead in Fuel, cc	Fuel Consumption, 300 cc, min	Intake Manifold Vacuum, in. hg Start	Intake Manifold Vacuum, in. hg End	Exhaust Back Pressure, in. hg Start	Exhaust Back Pressure, in. hg End
2	1941a	184	240	....	1.59	0.27	0.69	−0.36	0.89	....	104	....	30	....	1.9	....	....	....	....	....
	1941b	184	221	....	1.46	0.27	0.63	−0.36	0.56	....	91	....	18	1235	1.9	1.215	....	....	....	....
3	1941	178.5	215	0.37	0.68	0.35	0.66	−0.51	1.0	9.5	90.7	5.6	22.4	240	1.37	1.19	9.7	....	1.4	....
4	1941	178	226	0.46	1.95	0.31	0.82	0.05	.60	....	39	....	5	775	....	1.132	12.2	11.8	1.8	1.5
6	1941	174	209	0.46	1.53	0.3288	0.8706	Alk	0.50	8	93	3	32	955	1.2	.89	14.2	12.7	.92	.75
8	1941	177.6	222	0.44	1.33	0.32	0.62	Alk	1.32	52	104	7	24	804	....	....	....	....	....	....
9	1941	178	208	0.32	1.10	0.30	0.64	−0.50	1.25	0	40	0	4.0	344	....	1.25	....	....	1.1	....
11	1941	176.7	227.7	0.375	1.335	0.293	0.757	0.482	1.62	3	79	1	2	720	1	....	13.1	12.2	1.4	1.4
12	1941	....	....	....	....	....	....	....	....	....	....	....	....	450	....	1.15	14	14.2	....	....
13	1941	176:3	233.4	0.46	1.48	0.33	0.664	−5.3	0.95	....	50.2	....	24.8	802	1.24	1.25	12.3	12.4	1.0	1.0
16	1941	....	....	....	1.12	....	0.56	....	0.73	....	77	....	7	510	2/2.5	1.365	14.3	14.1	....	....

Note: Test a, oil temp. 280F; Test b, oil temp., 265 F. These two tests were run to show effect of oil temperature.

Laboratory	Engine	Air Fuel Ratio Start	Air Fuel Ratio End	Spark Advance, deg BTDC	Ring Wear, in. No. 5 Piston Top	Ring Wear Middle	Ring Wear Bottom	Bearing Loss, g Top	Bearing Loss Bottom	Bearing Loss Average	Engine Rating, % Varnish	Engine Rating Sludge	Engine Rating Total	Oil Pressure, psi	Sulfur in Fuel, %
2	1941	14.2	14.0	..	.....	.....	.....	0.21 0.20	0.22 0.21	0.215 0.205	....	....	....	....	0.06
	1941	15.0	15.0	60	0.0023	0.0025	0.002	0.17 0.16	0.16 0.17	0.165 0.165	80	77	78.5	....	0.06
3	1941	16.5	....	50	0.040	0.036	0.047	0.0986 0.0745	0.0710 0.1000	0.0848 0.0873	95.5	69.5	86	15.9	0.039
4	1941	14.2	14.0	52	0.002	0.002	0.003	0.138 0.165	0.159 0.228	0.1485 0.1965	99	97	98	12/12.5	0.03
6	1941	12.4	12.5	..	0.005	0.003	0.009	0.2406 0.2108	0.1583 0.1628	0.19945 0.1868	81.6	66.6	76	16	0.043
8	1941	....	....	..	0.010	0.009	0.014	0.089 0.098	0.109 0.089	0.099 0.0935	99	83	92.5	....	....
9	1941	13.7	....	52	0.021	0.021	0.023	0.1575 0.1256	0.1024 0.1170	0.12995 0.1213	96.3	83.7	91.1	....	....
11	1941	14.0	14.3	..	0.002	0.001	0.004	0.067 0.097	0.101 0.073	0.084 0.086	98.6	96	96.3	....	0.0517
12	1941	14.5	12.9	..	0.005	0.004	0.008	0.1672 0.1573	0.2268 0.1757	0.1970 0.1665	99.1	83.4	91.1	....	....
13	1941	15.0	....	..	0.004	0.005	0.005	0.27 0.25	0.29 0.27	0.28 0.26	91.25	97.25	94	12-15	0.035
18	1941	14.6	....	55	0.003	0.003	0.004	0.119 0.120	0.119 0.114	0.119 0.117	92	91.5	91.5	13	0.037

Table 4 – Cooperative Test Data on B-14 Oil
36-hr Results

Laboratory	Engine	V/100 New	V/100 Used	Con. Carb. New	Con. Carb. Used	% Ash New	% Ash Used	Acid No. New	Acid No. Used	Nap. Insol., mg/10 g New	Nap. Insol., mg/10 g Used	Chl. Sol., mg/10g New	Chl. Sol., mg/10g Used	Oil Consumption, mpq	Lead in Fuel, cc	Fuel Consumption, 300 cc, min	Intake Manifold Vacuum, in. hg Start	Intake Manifold Vacuum, in. hg End	Exhaust Back Pressure, in. hg Start	Exhaust Back Pressure, in. hg End
2[1]	1941	108	216	....	1.86	0	0.17	0.06	4.29	....	169	....	148	410	1.9	1.159	....	....	....	....
3[6]	1941	108	262	0.008	1.68	None	0.05	0.093	5.83	3.5	240.3	1.5	237.1	220	1.37	1.29	11.9	....	2.6	....
4[7]	1941	108	420	0.03	2.25	Nil	0.12	....	3.3	....	305	....	300	350	1.5	1.165	11.2	....	1.8	1.2
6	1941	108	391	0.025	2.18	0	0.0044	0.03	4.63	7.3	262	7.3	252	424	1.2	1.046	13.8	13.8	1.0	1.0
11[2]	1941	106.9	219	Nil	1.283	Nil	0.120	0.048	5.69	3	107	0.5	105	356	1	1.455	13.8	13.2	1.4	1.4
9	1941	108[8]	....	0.01	1.7	0.001	0.02	−0.025	7.8	0	177	0	175	318	....	1.263	....	....	0.7	....
12	1941	....	....	....	....	....	....	....	....	....	....	....	....	325	....	1.29	....	....	....	....
13	1941	108.3	233.5	0.01	1.40	0	0.014	0.005	3.70	....	153.9	....	153.9	167	1.24	1.29	11.0	12.5	0.8	1.0

Laboratory	Engine	Air Fuel Ratio Start	Air Fuel Ratio End	Spark Advance, deg BTDC	Ring Wear, in. No. 5 Piston Top	Ring Wear Middle	Ring Wear Bottom	Bearing Loss, g Top	Bearing Loss Bottom	Bearing Loss Average	Engine Rating, % Varnish	Engine Rating Sludge	Engine Rating Total	Oil Pressure, psi	Sulfur in Fuel, %
2[1]	1941	14.35	....	52	0.004	0.005	0.005	0.02 0.01	0.02 0.01	0.02 0.01	58	44.25	52	....	0.06
3[6]	1941	17.3	....	50	0.024	0.024	0.056	0.0585 0.0963	0.0635 0.0586	0.0610 0.0775	52.5	49	51.25	16.1	0.039
4	1941	14.2	14.0	..	0.002	0.002	0.004	0.026 0.008	0.013 0.006	0.0195 0.007	46.6	48.8	47.5	10/11	0.03
6[7]	1941	14.3	....	..	0.003	0.002	0.002	0.0327 0.0386	0.0266 0.0194	0.02965 0.0290	64.5	33.3	52	15/16	0.043
11[2]	1941	14.5	14.3	..	0.001	0.002	0.003	0.039 0.043	0.027 0.006	0.033 0.025	68.2	34.7	53.2	....	0.0517
9	1941	13.2	....	52	0.033	0.038	0.040	0.1378 0.0796	0.1166 0.0680	0.1272 0.0738	44.8	46.65	45.5	....	....
12	1941	14.3	13.6	..	0.001	0.000	0.003	0.0734 0.0605	0.0514 0.0554	0.0624 0.0579	43.5	18.2	33.7	....	....
13	1941	14.1	13.9	..	0.045	0.039	0.049	0.05 0.06	0.03 0.03	0.04 0.045	72	41.5	60	12-15	0.035

Notes:
1. Run stopped at the end of 19.5 hr.
2. Run stopped at the end of 33 hr due to varnish deposits.
3. Data are incomplete; balance will follow
4. Bearing weight taken at 26.5 hr due to excessive noise and pounding.
5. Run stopped at 8 hr so that data could be obtained for paper. Acid number was 10.
6. Run stopped at the end of 16 hr due to high acid number.
7. Run stopped at the end of 24 hr due to high acid number.
8. No used-oil viscosity due to clogging of Saybolt apparatus with sludge.

Table 5 – 36-Hr Oxidation Test Procedure

Briefly, this procedure involves the use of a current-model Chevrolet engine operated at 30 hp at 3150 rpm for 36 hr, with the water out at 200 F and the oil sump temperatures held at 280 F. The air-fuel ratio is held at 14.5 : 1. The fuel is "regular" gasoline, octane number, 72 minimum, sulfur, 0.04% max., lead content between 1.5 and 2.0 cc tel/gal. Full details of this test procedure will probably be made available by the SAE Committee in the near future.

Reference Oils

The reference oils were obtained from various sources by Dr. Floyd Miller and made available to the cooperating laboratories. The conventional oil inspection data, as supplied by Dr. Miller, are given below:

Oil	B-1	B-2	B-3	B-4	B-11	B-12	B-13	B-14
Gravity; deg API	24.3	30.1	29.1	22.5	25.5	32.0	31.1	25.4
Flash, F	425	475	480	415	375	400	400	325
Fire, F	475	535	540	455	390	440	450	360
Vis/100 F (SSU)	584	411.4	434.5	527	195	175	178	108
Vis/210 F (SSU)	60.2	60.4	60.2	55.5	44.1	45.5	45.5	38.6
VI	57	106	99	34	58	109	106	24
Color, R	2¼	9¼	6	15	2	12	6¼	12¼
Carbon Residue, %	0.29	0.11	0.46	0.04	0.25	0.02	0.45	0.02
Pour, F	−15	−15	−15	−25				

DISCUSSION

Relative Corrodibility of Oils Containing Various Acids

— C. J. Livingstone and W. A. Gruse

Mellon Institute of Industrial Research

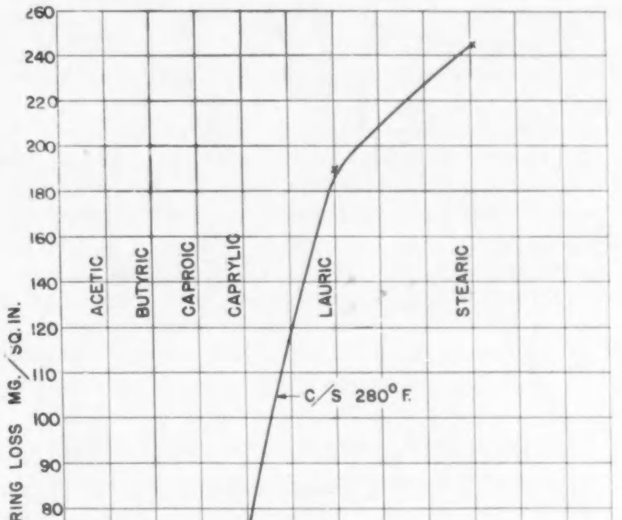
WE regard the paper by Messrs. Mougey and Moller as an excellent piece of work in the difficult field of harmonizing results of engine tests and in explaining variations in what is set up to be a standardized procedure. Their success was perhaps more evident in the paper by the same authors before the American Petroleum Institute at San Francisco in the fall of 1941. The fact that some of the data were at first in disagreement and are now in harmony furnishes good support for the validity of the method. The harmonizing process has brought out a good deal of new information. The material we wish to present may be of value in explaining some discrepancies.

It has been widely assumed that corrosion of alloy bearings is caused only by acids developed in the oxidation of crankcase oil at high temperatures. Chemists have known that such acids differ widely in properties. Some are as volatile as water and are strong corroding agents; others are hardly more volatile than the oil itself and attack metal less vigorously. L. L. Davis pointed out several years ago that bearing shells put in the air exhaust from an Underwood oxidation apparatus were often corroded by the acids formed from the oxidizing oil and evaporated in the air stream. We were interested in the effects which might be produced in engines when the character of the organic acid in the oil was changed. We, therefore, added to the crankcase oil in an experimental engine, one at a time, a series of organic acids, varying in character from the volatile and highly corrosive to the much heavier, less volatile and in general less corrosive. These changes are indicated by the number of carbon atoms contained. The acids can be regarded as resembling those likely to occur in used oil. Each acid was added to the oil to a neutralization number of 1.0 and tested with Cu/Pb and Cd/Ag bearings at 180 and 280 F oil temperature for 5 hr at full load. The results are shown in Fig A. It can be noted that over the entire range of acids used, Cd/Ag bearings were not corroded beyond normal wear loss, when the oil temperature was kept at 180 F. By contrast the same alloy was badly corroded by all except the lightest acids when the temperature was held at 280 F. The failure of the lower acids to corrode is explained by the fact that, though they are more active, they do not stay in the crankcase more than a very short time. The engine was well ventilated and bearing shells placed in the breather were corroded when these lower, more volatile acids were used. The higher acids, though less active, stayed in the oil, in spite of the ventilation and did more damage in the long run.

The results on Cu/Pb are somewhat different. Both at 180 and at 280 F, marked corrosion occurred; however, the lower acids did not corrode as much as the higher ones and again for the reason that they did not stay put.

To connect with what we wish to say next, it should be noted that the unburned heavy end of gasoline fuel, the diluent ordinarily encountered in cold-weather operation, is a likely source of lower molecular weight acids of this class,[a] since it has been oxidized by exposure to flame conditions. Such lighter acids could also be derived from the oxidation of the lubricating oil if temperature is sufficiently high. The heavier acids could not come from oxidation of diluent, and must be attributed to oxidation of the lubricating oil.

Since sulfur or sulfur compounds are practically always present in fuel diluent, and since sulfur is sometimes present in oil, perhaps because it has been dissolved from some types of filter cartridges, we studied the effects of sulfur and acids, alone and together. The results are shown in Table A. All refer to Cu/Pb and the oil temperature was 180 F. Sulfur alone (0.5%) in the oil caused no appreciable corrosion or weight loss in 48 hr; the bearing was black and polished. Adding a heavy acid – C_{18} (stearic) – caused only small bearing loss and only a trace of bearing growth. It must be remembered that stearic acid without sulfur corroded the bearings rather badly. When a C_4 acid instead of a C_{18} acid was added, a very interesting thing happened. The bearings in each case grew in size and, if the original clearance was small enough, the growth continued to the point of contact. When this happened, either the bearing seized or some material was sloughed off and the bearing lost



■ Fig. A (Livingstone-Gruse discussion) – Bearing weight loss for various acids in oil at 180 and 280 F

[a] See *Industrial and Engineering Chemistry*, Vol. 19, 1927, pp. 312-318: "Acids in Automobile Crank Cases," by A. F. Meston.

weight. These results occurred in a very short time, and it must be realized that the treatment was drastic and the conditions severe. The content of acid is not unusual, but the percentage of sulfur is far higher than ordinarily encountered. The experiments are, therefore, accelerated ones. Similar action in full-scale engines would occur mostly on rubbed areas of local overheating, and loss of metal would proceed gradually rather than all at once.

It must be remembered, however, that sulfur is not always harmful. Small proportions of sulfur, present as certain compounds and in the right amounts, may be quite beneficial.

The Chevrolet tests discussed by Mougey and Moller are all made at 280 F oil temperature and should correlate well with service in which the temperatures and bearing loads are comparable. Much corrosion of Cu/Pb bearings appears to occur, however, in fleet service involving low temperature and intermittent loads. We suspect that our preceding results are related to this latter phenomenon. Some of the characteristics of this type of corrosion are given in Table B. This trouble has been relieved or eliminated by use of some or all of the expedients given in Table C.

Table A – Copper-Lead Bearings – Crankcase Oil Temperature – 180 F

Test Oil	Duration of Test, hr	Bearing Loss mg/sq in.	Original Bearing Clearance, Thousandths	Decrease in Clearance Due to Bearing Shell Growth, Thousandths	Remarks
Oil + Sulfur	48	0.5	1.2	0.2	
Oil + Sulfur + C_{18} Acid	10	0.5	1.2	0.2	
Oil + Sulfur + C_{18} Acid	15	10.4	2.5	0.2	Oil changed every five hours introducing new acid and sulfur
Oil + Sulfur + C_4 Acid	3¾	48.1	1.2	1.6	Seized
Oil + Sulfur + C_4 Acid	½	...	1.2	1.2	Bearing tight
Oil + Sulfur + C_4 Acid	5	+0.5	2.5	2.0	Bearing just free

These data indicate that the Cu/Pb bearing, unlike the Cd/Ag bearing, is subject to a variety of attack which is not dependent on oil oxidation or on heavy-duty operation, but may occur under conditions of ordinary duty. This produces a rather confusing situation from the standpoint of oil testing.

Table B – Conditions Under Which Corrosion Has Occurred

1. Heavy-duty engines of good design; oil temperatures not above 180 F.
2. C/S bearing in same engine at same time not corroded.
3. Occurs in city service rather than in long hauls.
4. Pitted timing chains observed along with C/L bearing corrosion.
5. Occurs most in recently overhauled engines.
6. Bearing clearance suspected of being too small.
7. Copper sulfide identified in corrosion products left on bearing surfaces.
8. Has occurred with well-inhibited oils.

As we have pointed out elsewhere, a standardized multicylinder engine test is highly useful so long as the conditions are set to give proper correlation with service. When service conditions change, however, a new set of test conditions and a new correlation are necessary; this is sometimes forgotten and causes much confusion. When service and multicylinder testing do not agree and the reasons for the disagreement are unknown, a more fundamental kind of study is required. For such problems we have found the single-cylinder and prototype engines most useful. Because such equipment has not been on the market in the past, we have designed and are building such engines for use in our own organization. We are convinced of the utility of both the multicylinder and the single-cylinder engine, each for a specific purpose.

Table C – Means of Preventing Low-Temperature Corrosion

1. Proper fitting and machining of bearing shells.
2. Frequent oil change during break-in, to avoid formation of oil acids which might corrode bearings before protective films are formed on bearing surface.
3. Avoiding too drastic cleaning of engine at overhaul for same reasons as under 2.
4. Provision for good engine ventilation and avoidance of dilution.

Points Out Dangers of Oversimplification

— L. Raymond

Tide Water Associated Oil Co.

SIMPLIFICATION of lubricant testing is a very desirable step, particularly at the present time when the need for heavy-duty lubricants is increasing. However, it is a step which must be pursued very cautiously if the danger of over-simplification with resultant unsatisfactory selection and grading of lubricants is to be avoided. Our own experience has been that lubricant reactions are rather complex and that oils respond relatively differently to changes in test conditions in any given engine. The complexity of results obtained on a number of different engines adds further uncertainty to the question of whether a limited testing schedule can give a sufficiently comprehensive picture of the relative deterioration or corrosion tendencies of current oils in all types of heavy-duty engines and service.

As the authors have indicated, the original Chevrolet test (now the 36-Hr Oxidation Test) was of 66 2/3 hr duration, the equivalent to 4000 miles, whereas the present modification is 36 hr. Results obtained in this laboratory in the earlier 66 2/3-hr test at a crankcase temperature of 280 F did not correlate very well with performance in the General Motors Diesel 500-hr test, and it is doubtful if tests of shorter duration will result in improved correlation. The fact should be noted also that a gasoline engine is being recommended to evaluate oils which will be used primarily in diesel engines and that, to date, only four brands of oils have been used for such evaluation.

Other work done by this laboratory on the effect of time and oil temperature in both gasoline and diesel engines with different types of lubricants indicates that it is unsafe to shorten a test by increasing the severity, due to the distortion of results which frequently occurs. There appears to be a tendency in the paper to set up different orders of corrosion resistance. The nature of the bearing corrosion process makes such differentiation very dubious where the data are based on one bearing weight loss value at one test condition. Many bearing corrosion curves are exponential and show a more or less definite induction period, followed by a rapid increase in the rate of bearing weight loss. It is necessary, for an oil to be considered safe under a specific condition of operation, that the induction period be not passed. However, when the induction period is exceeded, the corrosion rate increases so sharply that a large scattering of weight loss values for oils of the same corrosion tendencies can be expected due to minor differences in time or operating conditions. This was apparently the case with oil B-2. It is quite hopeless, therefore, to attempt to average values on oils unstable or corrosive under conditions of test, in order to establish the different levels of instability. It is necessary, rather, that runs be made at at least two different levels of a significant operating variable, such as oil base temperature, measuring the oil and bearing changes periodically during the course of the test until the induction period is passed or the maximum test time is exceeded. In this way a more usable measure of oil performance characteristics can be obtained for the determination of safe operating conditions. The criteria then become the limiting conditions of safe operation for a particular product, which approach conforms more closely to the manner in which products are viewed in the course of their development for field service. In view of this condition, it is felt that a radical revision of the present testing program must be effected if fruitful results are to be forthcoming.

Automobile ENGINEERING ORGANIZATION and PROCEDURE

by C. R. PATON,* R. F. KOHR,** and M. A. FORESTER†

THIS paper presents the results of a survey of the engineering organization and procedure of seven typical automobile manufacturers in the Detroit area made by the authors at the request of the SAE Passenger Car Activity Committee.

Fifteen general conclusions are presented by the authors as a result of their studies of the engineering organization and operating procedure of each of the seven companies. Important among these conclusions are the following:

- Organizations, especially smaller ones, tend to evolve around available men.
- None of the organizations studied is truly represented by its organization chart.
- Design and experimental functions cannot be completely divorced.
- The horizontal type of organization is better for successive continuing programs, such as yearly automobile models; and the vertical type, for non-successive continuing programs.
- The smaller the company, the greater the need for engineering cost control.
- An adequate system of test records is an economy.
- The committee method of reaching decisions on major design questions has many advantages.
- The dual dollar and time budget method of project control seems highly advantageous.
- A full combination of production and engineering might be advantageous.

THIS paper was prepared at the request of the Society of Automotive Engineers' Passenger Car Activity Committee, as a cooperative undertaking. It represents a general survey of the organization and procedure of several typical automobile engineering departments in the Detroit area. A complete study of only one of these organizations would embrace ample material for a complete paper on this subject. A sizable volume of valuable material would result from reasonably complete description and study of the engineering organizations and procedures employed by all of the automobile companies.

We already have one excellent paper[1], completely covering this subject for one large car and truck builder, and it is desirable that we eventually have similar complete coverage of other automotive organizations, both large and small. Thorough analysis of this field is obviously outside the scope of this survey. Rather, we must restrict ourselves to a general and brief review of seven typical readily accessible organizations in the Detroit area, which have kindly volunteered their cooperation in this joint effort. As has been customary with similar committee-sponsored papers of this type, company identifications have been excluded purposely and cooperators remain anonymous. The authors, however, wish here to thank all those who assisted in making the necessary information available to us. All companies were not contacted. Those asked to contribute were chosen arbitrarily on the basis of accessibility or spe-

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[1] See SAE Transactions, November, 1938, pp. 455-464: "Experimental Procedure of Testing and Organization," by J. M. Crawford and P. A. Collins.

THE AUTHORS: CLYDE R. PATON (M '23) had been chief engineer of Packard Motor Car Co. for ten years at the time this paper was written – and had been associated with the Packard engineering department since 1930. Previously, he was a member of Studebaker's research department. During the last war, Mr. Paton enlisted in the U. S. Army Air Service and following its conclusion became student assistant in mechanical engineering at University of Michigan and later assistant mechanical engineer with the NACA before joining Studebaker. He is now executive engineer, Allison Division, General Motors Corp. He is a past vice president of the SAE and a past chairman of the SAE Detroit Section.

ROBERT F. KOHR (M '20) was experimental engineer with Packard when he collaborated on this paper and has since joined the Product Studies Division of General Motors Corp. Prior to his work at Packard, Mr. Kohr was friction materials engineer with the Thermoid Co. of Trenton, N. J., and earlier had been chief engineer of Asbestos Mfg. Co. Mr. Kohr was an Engineer Corps and a Tank Corps officer in World War I. A University of Michigan graduate, he worked at the Bureau of Standards, the Studebaker Corp., and Bendix Aviation Corp., before joining Asbestos Mfg. Co. in 1935. M. A. FORESTER is business manager of the Packard engineering department.

cial organizational interest or other reasons. Some were omitted from this survey due to the lack of time for 100% inclusion, or because they were substantially covered by others included. The authors admit their inability within the limitations of a paper presentation adequately to analyze and summarize all of the material obtained from the several cooperators. Points of similarity and difference, however, are brought out, together with mention of several of the more interesting procedures being followed by the different companies. Finally, we attempt to reach certain conclusions and recommendations, which occur as a result of this survey. Conclusions in some cases are those of contributors; in other cases, those of the authors.

It should be borne in mind that most of the information included in this paper applies to engineering organizations operating solely to further the engineering and production of civilian motor vehicles. Full allowance could not be made for the changes that already have occurred in the transition from civilian to war production.

We proceed to a brief description of the organization and interesting operating procedure of each of the seven companies to which we have previously referred.

■ Company A

This company produces a large number of different designs at different plants. The engineering organization is consolidated and its management is highly "central" in character. Because of the consolidation of engineering functions into a single unit, a high degree of product interchangeability is obtained and a uniformity is secured in product design character, performance, feature appeal, general standard of design and manufacturing. Centralized engineering brings the full force of available talent and development facilities to bear uniformly and effectively on the complete line of products.

This engineering organization includes about 1800 persons. Design direction and product character are determined by engineering representation within the general management group and by a committee type of engineering direction extending from the top well down into and throughout the organization. This organization represents one of the most successful and interesting plans of engineering procedure examined. Experience has shown that, although research function personnel must be relieved of routine production matters to be effective, best results are obtained by retaining research as an internal rather than an external group. Research programs are developed by what in effect is a "research council," members of which are attached to the several separate developmental groups. Thus, research in this organization is kept inside of, and under the control of, the various divisions. With this arrangement research can be kept separate from routine production problems, but also kept from getting too far afield from production possibilities.

Design and experimental functions in this organization are fully cooperative under the top directive committee, but are independent in such a way that no designer or experimental man can force through any particular favored design. Agreement between designer and experimental engineer is required, otherwise the question must be referred upward for review by higher authority. This arrangement makes possible a major spreading of design and development responsibility without hazard of improper authority delegation and thus avoids necessity for overloading the "chief engineer" who, in this case, is the top directive committee.

A project system is used, which assures follow-up and reporting of all work done. It is felt that this operation is particularly necessary because of the size of the organization and the spreading of responsibility among the personnel. Of special interest is the fact that every project is controlled by budget as to both time and expense. When the end of either the pre-assigned time or expense is approached, the individual responsible for the project is notified by the central accounting system and must ask for more time or money, or both, and justify his request. This dual check system has been very effective in preventing projects from dying in the design room or in the laboratory.

Engineering cost study and central planning functions, although part of the manufacturing organization, are so located and operated as to be effectively available as a part of engineering.

This company employs extensive laboratory facilities for checking out separate design units, thus avoiding necessity for excessive road testing of complete units which would otherwise be necessary. Extensive road tests and cross-country work, however, are included for checking both preliminary and final designs, it being agreed that laboratory test procedures cannot safely be substituted entirely for road tests. Emphasis is, however, on laboratory rather than on the road test, because it is recognized that man's limited senses can best be amplified and made quantitative by instrumentation – frequently only possible in the laboratory.

The laboratory proof precedes road tests, reduces their extent, and cuts down their cost. With establishment of additional correlation between road and laboratory, this company's trend is toward further increase of dependence on the laboratory test procedures.

Cross-country testing is employed as often as appears necessary and, in addition, information is secured from the operation of fleets in the hands of users.

Production engineering is handled through the chief engineers of the several manufacturing divisions and their assistants. These plant chief engineers maintain no separate product design facilities and do not make production releases, although they have full authority to make emergency changes to keep production going. They are the "front" men for their divisions in cases of sales and service contacts and perform all engineering functions necessary to continued manufacturing, including follow-up of service complaints, contact with other divisions, with the sales department, and with central engineering.

■ Company B

This engineering organization serves one division of a corporation and has to do solely with the products of this division. Operating independently, individuality of product is preserved and engineering along independent lines is encouraged.

With a personnel of about 550, this group provides all the engineering effort for an extremely large production. With this fact in mind it is not surprising that little attention is paid to the *engineering cost* of arriving at a reduced *production cost.*

Design work in this organization is guided by a project engineer assigned to the job through a group leader in

the drafting department. There appears to be little tie-in between design and experimental as, when a unit is turned over to the experimental division for building and test, it is beyond the direct control of the project engineer except as he receives information through progress reports or direct information concerning failures or unexpected performance characteristics. In this case he has an opportunity to inspect the part or unit in question and observe for himself the nature and extent of the departure from expected performance.

In order to make sure of written records in this rather large organization, no work can be started without a written design order, building order, or test order. Once started such an order is followed up automatically through an adequate clerical force.

It is noteworthy that neither time nor money is budgeted on any job. If a job is regarded as necessary, it is followed to a conclusion regardless of time or expense. In exceptional cases, of course, a project may drag along until interest in it is lost and then be canceled on account of accumulating cost.

A central research organization is regarded as an accessible convenience. When a definite project can be handled more advantageously by this group, a work order is issued and time and material charged against such order. The research group may suggest to engineering heads, but has no jurisdiction. Its consultation service is, of course, always available.

A test staff is maintained at the proving grounds available to this company. It is interesting to note that, while both laboratory and road-test facilities are of the best, neither is emphasized. The nature of the test in question determines the facilities to be used and the selection is always on the basis of suitability. One example of an investigation best carried out in the laboratory is an axle seal comparison, in which case consistent road-test results could neither be obtained quickly nor without the use of a large number of cars. Laboratory results are always checked automatically as soon as the units involved are installed in the road-test cars.

In the case of a device used to replace an existing unit of known performance and with some laboratory test background, it is obviously quicker to check the new part or unit by laboratory test. Where it is a new device with no basis for laboratory performance, it must go immediately to vehicle installation in order that its operating characteristics may be learned.

In addition to routine endurance tests conducted at the proving grounds, cross-country test trips are undertaken usually twice a year under the direct supervision of the chief engineer, assistant chief engineer, or experimental engineer. The fall trip is made with preliminary development cars accompanied by a current model for comparison. The spring trip is a check on the car as actually released, representing the final design, and serves to re-check features changed as a result of the fall trip, as well as sometimes uncovering other faults not previously encountered.

Production engineers are stationed at each manufacturing plant, and all engineering contact with the plants are through these men. They are responsible to the chief engineer on technical matters, but to the plant manager on matters of policy. After performance standards are established and a design released, the selection of materials and contact with sources is in the hands of production engineering.

Two men in the engineering department act as service engineers and see that all service complaints get to the proper project engineer for solution. If sufficiently prompt or satisfactory action cannot be secured in this way the service engineer may go to the assistant chief engineer to make sure that an answer is forthcoming.

■ Company C

This engineering department serves one of the smaller divisions, with respect to production volume of a large corporation. In common with other divisions, it is unhampered as to maintaining its own individuality, with the attendant penalty of having to produce engineering results with a comparatively small group.

In this group of about 320 people the division of responsibility is almost completely vertical. Under the chief and assistant chief engineers, the duties of the department are split between department heads having under their control rather clean-cut sections of the job.

These departments, five in number, fall under the headings of – engine, chassis, body, electrical, and technical data. Each department head is responsible for the units under his department from the start of design through building, test, production, and service. While putting a heavy load on each department head, this policy ensures that all the information on each subject is included in the experience of a single department.

Another group of departments, known as "service departments," includes the garage, drafting, specifications, laboratory, and administration and procurement. These departments are charged with doing the work laid out by the engineering department heads. A group of floating project engineers, known as the research division, is at the disposal of the department heads and its personnel may be assigned by them to any project as occasion demands, although the tendency is to use them on future developments.

Each department head has a few assistants who tend to become specialists but who are available on any problem arising within the department. The average department consists of 8 or 10 engineers.

In general, the service departments are not directly under any engineering department head, although the electrical laboratory is an exception and falls directly under the electrical engineer. Avoidance of specialization in the service departments helps to keep the men busy and avoid low spots.

Project appropriation requests covering the larger and more general jobs must be signed by a department head and approved by the chief or assistant chief engineer. Such a request carries a cost estimate which is broken down among the departments which will do the work. It is assigned a work order number which is then applied to each test request coming under its general scope.

Test requests carry no cost estimate, but do carry a required completion date which must be agreed to by the service department doing the work. The request must be approved by the head of the technical data department, who represents the chief engineer in this respect. It is subsequently followed up as to completion within the allotted time.

Reporting of investigations is principally at the discretion of department heads who see that reports are prepared if they believe it desirable. Very small jobs are probably

not reported at all, and no follow-up system is in force to ensure preparation of reports.

No test order is required for a job requiring less than one day's work, and each department has a general project covering miscellaneous work, and budgeted as such.

As in the other divisions of the same corporation, the central research group is employed when its facilities or experiences make such action economically expedient. Research as such is not emphasized within the division.

This company maintains a garage at the proving grounds, with a permanent foreman, but with cars operating out of the home garage and going in and out between shifts. No emphasis is placed on either laboratory or road test, suitability to the job in hand being the governing consideration.

There is no general policy on cross-country trips, although these trips usually are made two or three times a year under the supervision of a floating project engineer, who has developed as a "judgment expert."

Under this set-up the chief engineer and assistant chief engineer are concerned chiefly with policy and with design from an appearance standpoint. They receive all information directly from engineering department heads without the necessity of reading a mass of routine reports. The department heads carry *both the responsibility and authority*[2] in design and manufacture of the units under their supervision.

■ Company D

This company is another division of a larger corporation, and handles the engineering incident to fairly large production with a force of about 230 men.

This engineering organization tends to be vertical in function. Each engineer responsible for a given unit, as engine, chassis, and so on, directs the draftsmen in design, recommends experimental procedure, and functions also as production engineer and service engineer. He reports to the assistant chief engineer in charge of design.

The experimental division, also under an assistant chief engineer, receives experimental units and proceeds with test work, keeping the responsible engineer informed by reports, and advising him immediately and directly of anything unusual or unexpected turning up during test. It is only during the experimental work that the engineer responsible for the unit or part in question is relieved of direct responsibility.

Work orders issued at the request of the responsible engineers, authorize installation and insure follow-up and reporting of performance of experimental parts.

No budget on either time or material is applied to projects, although cost is watched to keep the departmental budget within bounds. In some cases when a record of expense on a particular project is desired, a work order is issued by the accounting system and this serves to accumulate the cost items as they occur.

Orders on design are mostly verbal, although on a new set-up the chief engineer issues a memorandum to the engineers involved, giving them authority to proceed. While the general projects are authorized by the chief engineer, authority to redesign, correct, and improve is in the hands of the engineer concerned. The general idea is cooperation and short cuts.

Because of the large volume of relatively unimportant matter, no written reports on experimental results are sent to either the chief engineer or his assistants, although complete records are secured and maintained by the technical data section under the experimental division.

There is no research division as such within this organization. Such research as is done tends toward development and bears on fairly immediate use.

As in the case of other divisions of this corporation, a proving-ground garage is maintained and all road test work is done there. Laboratory tests usually are paralleled by proving-ground tests although, in some instances, only one of these facilities may be employed.

Cross-country test trips are sometimes used although much similar information is gained from the use of the test cars during week ends and vacation periods by the responsible engineers.

■ Company E

This engineering department serves a large independent. No exact figure as to the personnel employed was obtainable but, if all activities, including road-test drivers, are included, it probably is somewhere near a thousand.

The organization tends to be horizontal rather than vertical, that is, the division of function appears to be between design, experimental, and so on, rather than between units or characteristic vehicle divisions, as motor, chassis, electrical, and so on.

Any major project is started by the chief engineer, usually on the basis of consultation with other top executives. It may result from a suggestion of anyone within the engineering department but, if of any size or importance, it will be judged by the chief engineer in the light of consultation with management.

On initiation of a new project, it goes first to an experimental design group where preliminary sketches are made. If these sketches look promising, a wooden mock-up may be made (as in the case of a new model car), and the mechanical design checked.

The experimental models are then built in the experimental machine shop. During this period the work of all experimental design groups is followed by a contact and follow-up engineer, who handles contact with the shop and the ordering of experimental parts. All changes go through his hands and he is responsible for coordination of deliveries so that all parts will be available for assembly at a specified time.

Production men from any department to be involved in manufacture are invited to examine the experimental parts from a production standpoint, and sources are checked for producing ability and material supply.

When the new unit is ready, it is run on a test course under a special schedule laid out for the testing heads at the direction of the chief engineer. During test, progress is followed by all the specialists involved, as carburetor, electrical, spring, sound, and so on. The type of procedure is agreed upon between the design and test heads.

At no time is any forecast made of the cost of a project, nor is any cost record kept. The management believes that the *engineering is necessary* and that the money has to be spent. Time is budgeted roughly as a reasonably accurate completion date must be met. Where it appears to be more suitable, units may be laboratory tested.

A separate research department is maintained, under the direction of the chief engineer, and is engaged chiefly on

[2] No procedure has yet been found which successfully separates authority from responsibility.

investigations somewhat off the beaten path, although any problem may be handed to the research staff, perhaps bearing on immediate experimental models or production.

Cross-country testing goes on the year around. There are probably 200 or 300 cars and trucks on the test roads or in various parts of the country in dealers' hands for test all the time. A number of caravan drivers are maintained whose headquarters are at the factory, but who spend most of their time on cross-country test trips wherever required.

Of interest with this company is the fact that production engineering is assisted by the inspection staff, which must not merely inspect, but also must help to correct.

■ Company F

This company is a well-known independent making several models. The division of engineering responsibility tends toward the horizontal, that is, preliminary design, experimental, production design, rather than the vertical type.

Any new development is first turned over to a designing engineer heading a research division, including a test engineering group. The unit is then built up in the experimental machine shop coming under this designing engineer. The chief test engineer carries through road test and is responsible through the designing engineer to the chief engineer.

If the job is satisfactory, it is turned over to experimental engineering and there changed, corrected, or otherwise made suitable for production.

From this point it goes to chassis engineering where the chief engineer carries on the design, releases drawings, and contacts vendors on engineering matters. He is responsible for all chassis production design. This division makes no prints, and no releases to the shop as such. The original drawings are released to a central planning division along with a bill of material and this division in turn makes prints and distributes them to the shop divisions involved.

Every item is estimated as to time, but there is no cost set-up in the engineering department. Overall cost is estimated, based on previous experience and the degree of change of product which is contemplated. No attempt is made to break this cost down into budgets for individual projects.

No research department in the usual sense exists in this company. The department bearing that name is actually engaged in preliminary experimental design.

While this company has no proving grounds, no overemphasizing of the laboratory results. The type of test most suitable is selected in each instance.

The records department under experimental engineering keeps all experimental records. Each test engineer is responsible for reporting his own test results. These reports are circulated only to the chief engineer through the experimental engineer. Additional contact with the chief engineer is obtained by a daily conference of one hour between the chief and the experimental engineer.

A service engineer within the engineering department receives all service complaints and runs them down with the proper department. He visits the field as engineering representative for the service department and works with the executive group on outside contacts.

There are no production engineers as such, this function being handled by contact men in the various departments under experimental engineering. Two men under the chief body draftsman provide the same service on body items.

■ Company G

This engineering group serves an independent company, which has made a practice of building a large proportion of its own parts and bodies and has required an engineering force numbering about 400. The organization structure is largely horizontal.

Projects in this department are authorized by the chief engineer, and an engineering records department keeps check of expenditures, notifying the originating engineer monthly as to expenses incurred. No time budget is established, but all projects are reviewed monthly to establish current priorities.

This company also has access to a proving ground and a good share of the development and experimental work is done there. Because of its readier accessibility many development projects are carried on by the use of cars and facilities of the experimental garage at the factory. There is no set rule and the facility which seems most suitable is used. A mechanical laboratory at the factory embraces the dynamometer, mechanical, and electrical laboratories, and is used for all endurance tests of units which can be accomplished by such means.

Reporting, both verbal and written, of test results is up to the engineer responsible for the work. An effort is made to see that every project, even though abandoned, has a brief report incorporated in its file in the experimental records division. Short cuts are the rule rather than the exception, and at times the only record of a test result may be in the form of a resulting production release.

A research department within the engineering department is made up of a research engineer and his staff which is used principally for analytic investigation of proposed designs, causes of failures, and also for some probing into the probabilities of new devices whose characteristics are experimentally unknown.

The division of duties in this department has tended toward the horizontal. Designs are usually worked out with frequent contact with the experimental engineering group. The latter, in turn, takes the lead during road test and development and works out the final changes necessary to complete the job.

Service complaints are handled by a service engineer who follows them up with the proper division of the engineering department.

Production problems are handled by a production engineer and his assistants, who are in constant touch with both production and engineering. He has the authority to adopt emergency measures to keep the line going, but always goes back to engineering for permanent change releases.

■ Conclusions

In our discussions with the various cooperating company representatives, we have reached certain conclusions which we include in closing:

1. We find generally that organizations have evolved around available men. This is especially true of the smaller companies who must put more dependence and greater individual load on key men than is necessary in the larger organizations.

2. None of the organizations studied is truly represented

(*Concluded on page* 464)

PRODUCTION TESTING FACILITIES

of Allison Division of General Motors

by H. J. BUTTNER

Allison Division, General Motors Corp.

THE author traces the background of present engine testing facilities with a note that the latest type of engine test stands are but elaborate developments of some of the first test stands used for Liberty engines.

Noted also in this paper are the considerations given to the factors involved in the choice of the basic design for the production stands.

Design features discussed include general building arrangement and construction, engine stands and mounts, operational equipment, temperature regulating equipment, plumbing, filters, weighing equipment, instrumentation, and sound treatment. The mount problem is discussed from the standpoint of design, vibration, and endurance operation. Methods for straightening air in U-shaped test cells are touched upon.

Layout of heat exchangers emphasizes rapid warm-up and economical water usage. Temperature control of glycol and oil is obtained by bypassing coolers. The plumbing is designed for ease of disassembly and cleaning. Full-flow filters have proved worth while in the oil system. Fuel measurement is obtained both by weight and by rotameters.

The operator's control layout is illustrated and discussed.

Sound - absorption problems are noted, and sound-absorption tests are illustrated.

THE AUTHOR: H. J. BUTTNER (M '40) has been in charge of all experimental testing of the Allison Division of General Motors Corp. since 1936. Starting his employment with Allison as test engineer, he was transferred in 1934, for two years, to the General Motors Research Laboratory in Detroit in the capacity of aircraft diesel engine project engineer. He then returned to become Allison's chief test engineer. Mr. Buttner received his B.S. in M.E. from Purdue University.

DURING the last 25 years engine testing facilities have kept pace with the development of aircraft engines. These facilities are as improved as the 2000-hp engine is over the 300-hp engine of the first World War.

A great deal of credit must be given to the pioneers of what has developed into a major portion of one of the most progressive industries of today. They had no background of thousands of hours of engine testing experience on which designs of test cells could be based. However, it is apparent that a correct basis of design was sometimes achieved. This fact is brought out in a close study of Fig. 1 showing a double stand and a common control room. This type is now the last word in modern design and is technically known as "straight-through" construction which is so essential in testing large engines with "flight"-type of propellers.

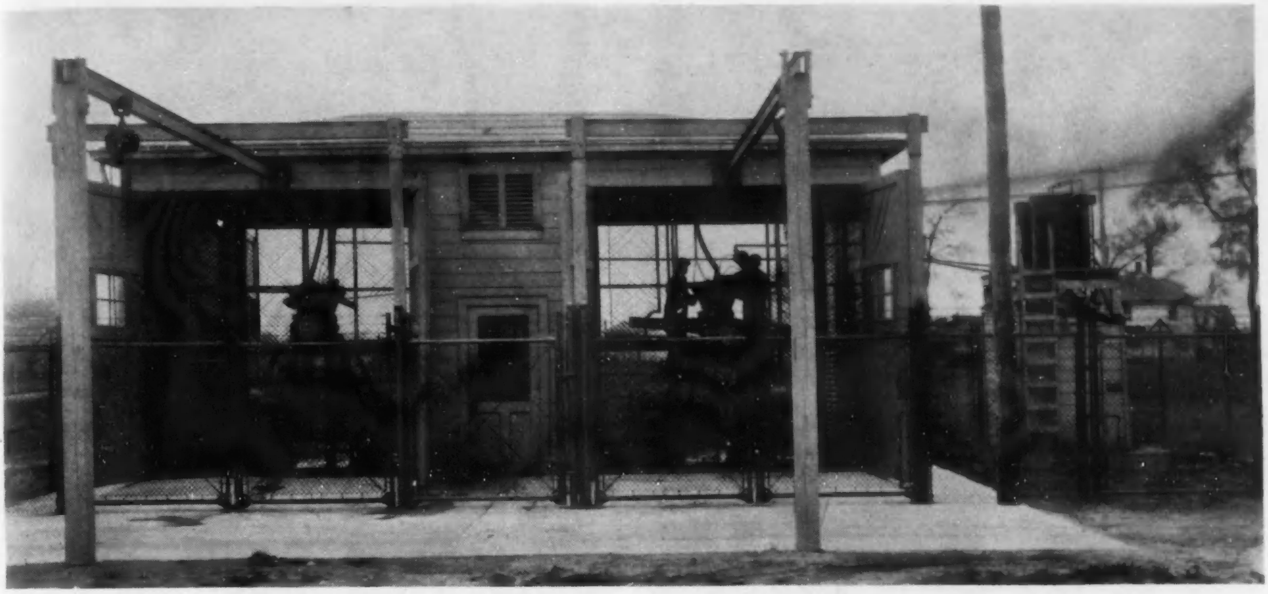
This particular photograph shows two of the test cells used for testing Liberty engines. The structure is of wood with no protection for nearby residents or for the stand operators. It is doubtful whether such a structure would withstand the beating imposed by air pulsations from a propeller on a 2000-hp engine.

Compare the equipment in this photograph with the equipment of today: concrete structures with walls 18 to 30 in. thick, soundproofed control rooms, soundproofed engine rooms, two or three thicknesses of safety glass protecting the operators, automatic temperature controls, instruments designed especially for the job, control desks resembling great theatre organs – requiring great skill and training to handle the multiplicity of controls, streamlined engine stands, controllable-pitch propellers, absorption dynamometers or generators which return the thousands of horsepower back through electric power lines. All of this shows the development resulting from thousands of man-hours devoted to the improvements which, in many cases, were absolutely necessary to keep pace with the increase in power output of engines.

When a newcomer to the aircraft engine production field is confronted with the problem of setting up testing facilities, a multitude of decisions have to be reached – a few of which are listed here:

1. How large an engine will have to be tested?
2. What different types of engines will be coming off the production line before stand obsolescence is reached?
3. How will the power output be absorbed?
4. Availability of cooling medium?
5. How much room is available for stands?
6. What amount of money can be expended?
7. When will the equipment have to be available to

[This paper was presented at the National Aeronautic Meeting of the Society, New York, N. Y., March 12, 1942.]



■ Fig. 1 – Liberty engine test cell

meet production schedules?

When Allison Division of General Motors entered the production field, the usual rush occurred to design and construct equipment and buildings. The ground rules, then laid down and rigidly adhered to since, specified that the greatest simplicity be maintained and that little deviation could be made from known practice.

The first decision reached pertained to the method of absorbing the power. Consideration was given to the use of electric dynamometers feeding back into power lines, to water brakes, to wooden test clubs, and to controllable-pitch propellers.

Controllable-pitch propellers were selected because it was felt that operating conditions would approach those of flight more closely. We felt that dynamometer operation would not show up defects in that the engine would not be subjected to lateral movements and torsional vibration as in an airplane. This belief was correctly proved by defects which appeared in engines tested with flight propellers after a quantity had been production tested on a dynamometer.

Other factors entering into a decision against dynamometer equipment included the relative ease of obtaining propellers instead of dynamometers, the ease of maintenance of propellers and their record of long life in aircraft operation, the difficulty of designing or obtaining couplings that could be relied on for continuous use, the great quantities of cooling water required for electric eddy-current absorption dynamometers or water brakes, and finally, the ease and simplicity of installation and operation of the engine.

The general arrangement of the test cells and equipment finally decided upon followed closely that used in the Experimental Laboratory. Minor modifications were made as warranted by production testing.

■ Building Arrangement and Construction

The first stands constructed were of brick and reinforced concrete. Brick was utilized to eliminate costly forms and expedite construction.

The general plan consists of pairs of engine rooms with a common control room between each pair. The control room and engine rooms lateral off a central corridor. Large double soundproof doors permit entrance to the intake end of the engine room. Single doors shut off general noise of the corridor from each control room.

The engine rooms are 20 ft square in cross-section and 80 ft long. The walls in the square throat are of reinforced concrete as a protection to personnel from possible danger should a propeller blade fail. For further protection the propeller is kept well forward of the operator's window and approximately in line with the fuel and oil weighing room.

The control rooms are 20 x 28 ft, exclusive of the previously mentioned oil rooms which project into the control room. A stairway in each room permits easy access to the basement in which are maintained all heat exchangers, filters, pumps, and plumbing.

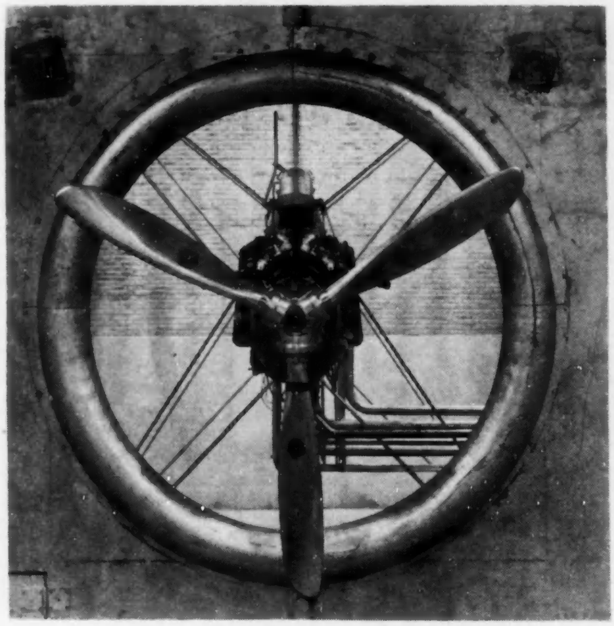
■ Engine Stands and Mounts

A great deal of development work has been done in attempting to determine the best type of mounting for production, as well as experimental, operation of the Allison engine. A decidedly different problem is encountered here as compared with that of flight operation.

In aircraft operation it is essential that the engine does not set up resonance in components of the ship's structure or instruments; also that the operation with respect to the crew be such as to minimize fatigue in long flights. This requirement generally requires extremely soft mounts capable of damping all modes of vibration.

In test-stand operation these considerations are not extremely important since the supporting structure can be made sufficiently strong to withstand most excitations set up by the engine-propeller combination. It is also possible to prevent vibration from reaching the operators' end of the controls by suitable designs.

The mount problem seems to boil down to the follow-



■ Fig. 2 – Propeller air-straightening plate

ing propositions:

1. It must not have resonant frequencies which can be excited by improper firing of the engine or propeller blade vibration due either to natural frequency of the blades (usually of too high order to cause difficulty), unbalance of propeller either static or aerodynamic, or forced vibration due to localized disturbance of the air flowing through the propeller.

2. It should not exert stresses on the engine due to relative stand movement. The engine should only be required to carry loads normally expected of it in operation for which it was designed.

3. It should be sufficiently soft to reduce impact loads.

4. Its design should be such that installation of the engine can be effected in the shortest possible time with a minimum training of personnel.

5. Interchange of engine models must be possible with fewest mount changes.

6. It must be capable of operation over long periods of time without replacements.

There are three methods of achieving the correct mount for a given engine type:

1. Mathematical analysis.

2. Vibration survey with engine-propeller-mount combination.

3. Endurance operation.

None of these methods should be used without consideration for each of the others. This rule has been proved rather conclusively in several instances.

In our operation we have found that natural resonance of the stand has not caused a fraction of the trouble that so-called "gust conditions" have. This statement is particularly true in U-shaped stands having vertical intake and exhaust stacks. In some cases where there has been continuous endurance operation, it has been necessary to resort to an air-straightener for propeller air. The usual circular throat track or tunnel was found to be very inefficient for this application unless a great length/diameter ratio of the throat is used, which design proves impractical in most stands.

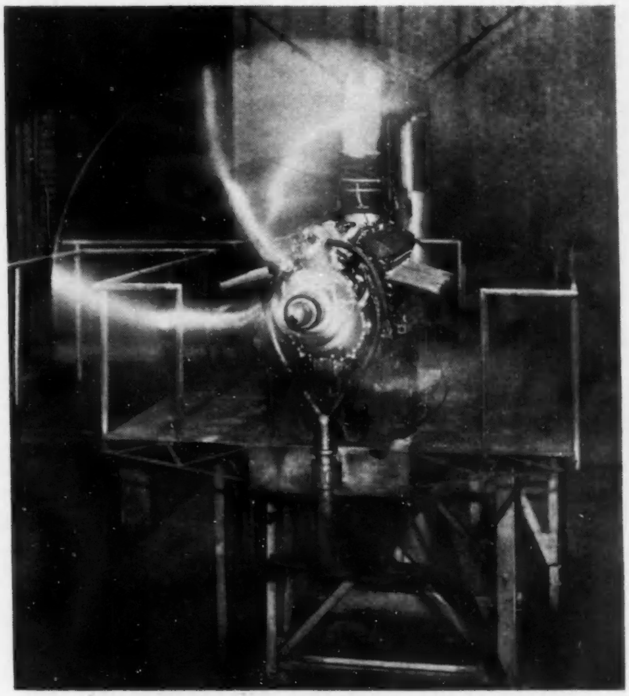
Fig. 2 shows a simple procedure which has proved highly satisfactory in straightening air. It consists essentially of a plate having a bore slightly less than the propeller diameter. This plate is located close to the trailing edge of the propeller. There is no back-flow of air in stands equipped with this device. The velocity of air through the cell is much greater than is the case when no such device is used.

Structural steel stands (Fig. 3) were first utilized in the test cells, but reinforced concrete stands are now used exclusively for all new construction. The latter proved better from many standpoints. Concrete stands are much easier to fabricate after the forms were available; they are easier to clean; and finally, due to the greater mass and rigidity, stand vibration in the operating range does not occur, resulting in low maintenance requirements. (See Figs. 4 and 5.)

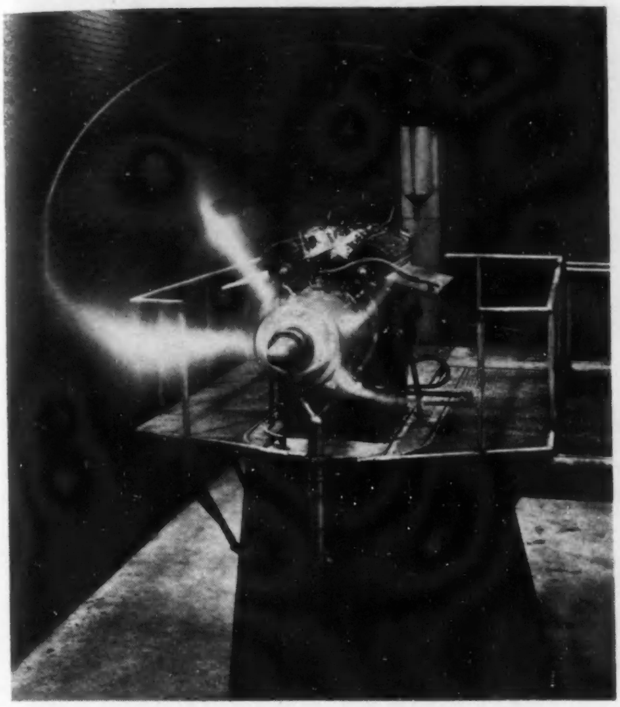
The engine mounts consist of cast-iron bases carrying rubber bushings at four points to which the engine is attached. These bushings carry torque and vertical and lateral forces through the compression of the rubber. The thrust forces imparted by the propeller are carried by shear loading of the same bushings.

Use of the rubber bushings greatly facilitates installation of the engine since there is little danger from distortion of the engine as would be true when bolting to rigid structures where mounting surfaces are not in a plane.

Fig. 6 shows a typical installation ready for mounting of the engine. In this picture can be seen the simplicity of attaching equipment which results in a great saving of time. This saving is brought out by the fact that it requires about 20 min to install or remove an engine with a normal crew.



■ Fig. 3 – Steel engine stand



■ Fig. 4 – Concrete engine stand

■ Operational Equipment

Heat Exchangers – Engine lubricating oil, coolant, and carburetor air temperatures are controlled through use of copper-tube heat exchangers. Cooling water is carried around the tubes of the glycol cooler and then through the tubes of the oil cooler. A thermostat controls the discharge water temperature. This procedure permits rapid warm-up of the engine and results in economical usage of water. The maximum temperature of cooling water permissible with extremely "hard" water is 140 F. For this reason it has been beneficial to bypass the fluid to be cooled around the heat exchangers for temperature control. Fig. 7 shows the glycol heat exchanger to the left of the picture. The carburetor air is heated in winter by one set of steam coils. Temperature regulation is achieved by controlling flow of water through a separate set of coils.

■ Temperature Regulating Equipment

As just mentioned, temperature regulation of the glycol and oil is obtained through control of the quantity bypassing the coolers. Valving is controlled from the operator's desk by positioning motors which regulate the position of the bypass valves.

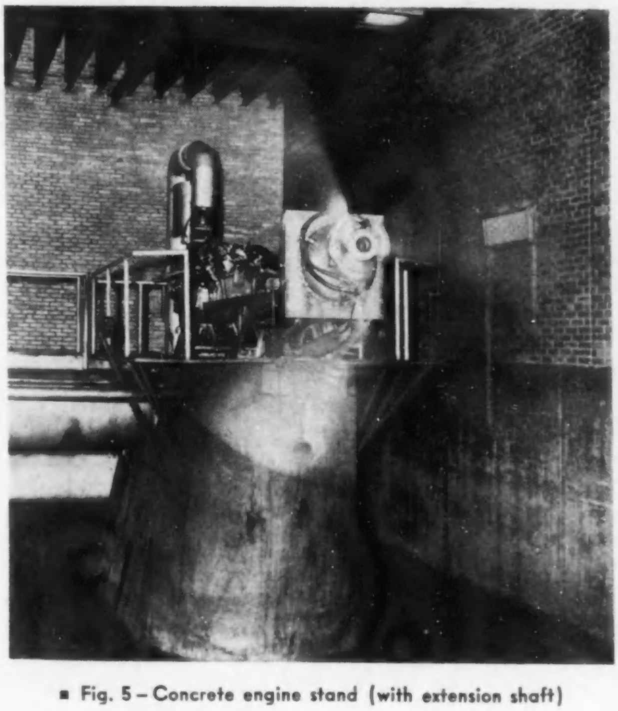
The position of the control motors is determined by the manual setting of the operator's positioning switches. No attempt is made to secure automatic control other than that of the discharge water valve which does not affect the operation of the cooler.

■ Plumbing

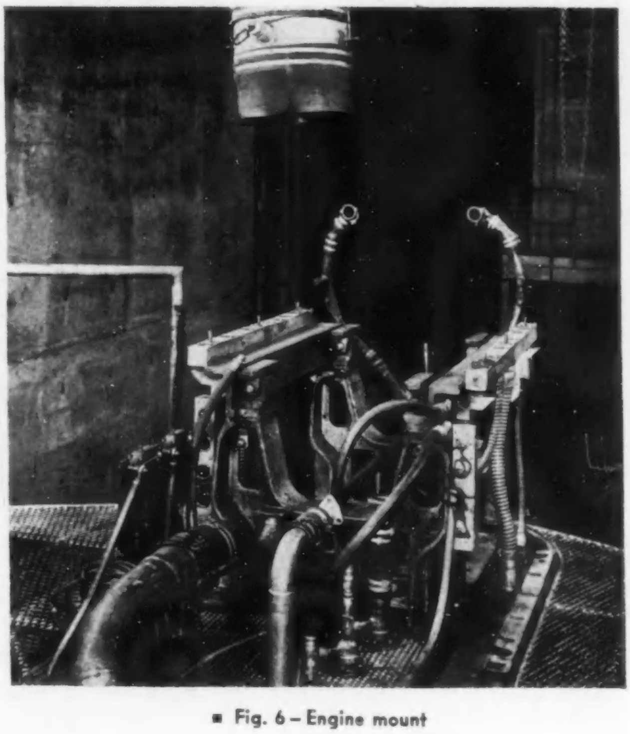
Great care has been taken to provide piping that could be easily disassembled and cleaned. Bolted flange connections are used throughout on both the glycol and oil systems. The oil system is composed of brass and hard copper pipe, and fittings having sweated joints.

The problem of dirt in the lubrication circuit has been responsible for the adoption of this type of plumbing. The care taken has been well worth while in that costly replacement of parts due to scratches has been reduced to a minimum.

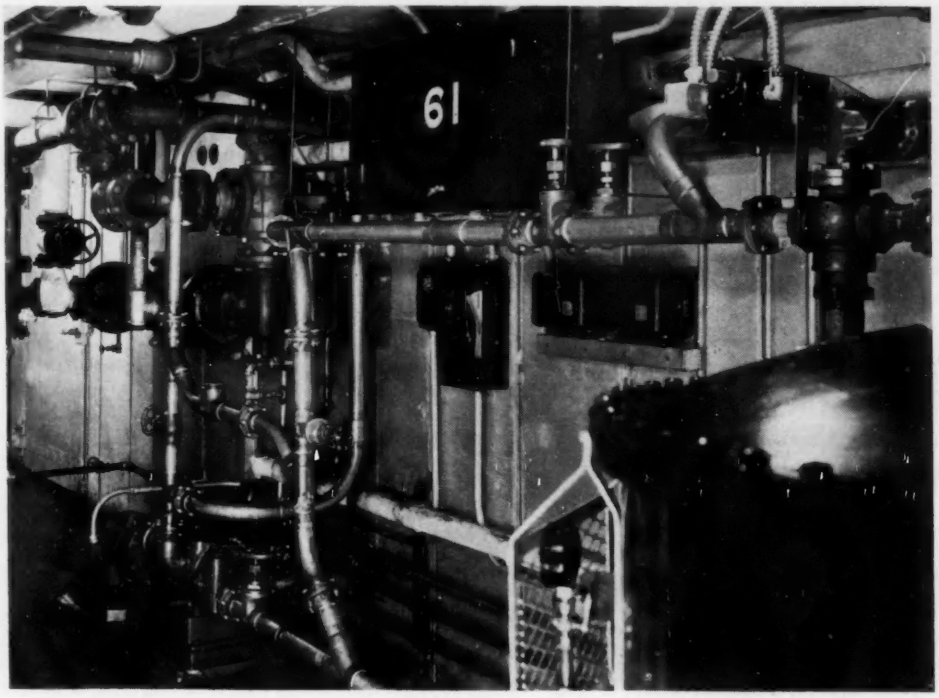
The lubrication plumbing seen in Fig. 7 seems to the casual observer to be quite complicated. However, when consideration is taken of the fact that two complete sets of lines are run to the engine in order to provide for



■ Fig. 5 – Concrete engine stand (with extension shaft)



■ Fig. 6 – Engine mount



■ Fig. 7 – Plumbing and heat exchangers

return of all engine oil to the basement after it has reached the engine, then passed through a full-flow filter, then back to the engine, through the engine and again back to the basement – there is justification.

■ Filters

The full-flow oil filters of either the centrifugal or Fullers-earth type have been an excellent investment, particularly in view of the possibilities resulting from the arrangement of the lubrication system. It is possible to circulate the oil in the entire system through the filters at high flow until analysis shows oil of a grade meeting laboratory requirements.

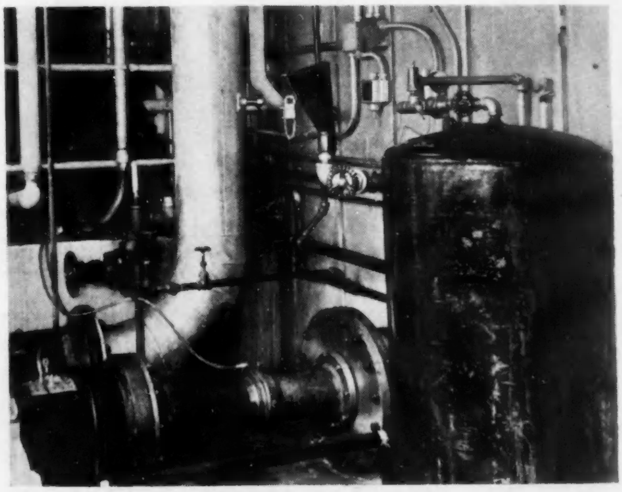
■ Air Filters

Operation of engines during building construction in the vicinity has shown the necessity for filtering of the carburetor air. Also the fallacy of carrying air through brick or concrete ducts after filtering was brought out vividly by rejection of engine parts subjected to this practice.

Two different types of filters are in use. The copper-wool unit impregnated with oil is fairly effective if the air velocity through the unit is very low and the units are given proper maintenance. The oil-bath filter, although in use a relatively short time, is believed to be more effective.

■ Weighing Equipment

Venturi meters (Fig. 8) were selected for measuring air flow to permit recovery of pressure loss encountered in fixed orifices or to eliminate the continuous changes required of smooth approach orifices to maintain pressure drops to a narrow range. With the venturi it is possible to use one size of throat for engines requiring maximum air flows varying as much as 30%, and yet to permit

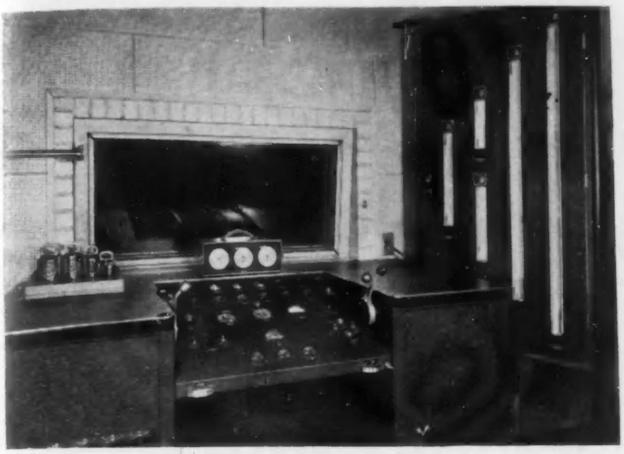


■ Fig. 8 – Air-measuring venturi

sufficient accuracy in the low cruise-power range.

The cost of venturi meters was kept low by use of a relatively short throat section accurately machined from brass castings and expanding cones rolled from sheet metal. Stand space was saved by slipping the expanding section of the venturi inside large pipes carrying air under the engine room floor. In this construction adequate protection must be given to all pipes to prevent rusting. Metallic zinc was sprayed on underground pipes as a preventive when paint treatment used on new installation proved unsatisfactory. A heavy galvanized coating is now used on all new piping.

Fuel weighing has long been a troublesome issue. Most accurate results are achieved by the known-weight system achieved by running out fuel from a tank mounted on postal-type scales and hanging a known weight on the



■ Fig. 9 – Control desk (old)

tank after the scale trips and recording the time required for the scale to trip again.

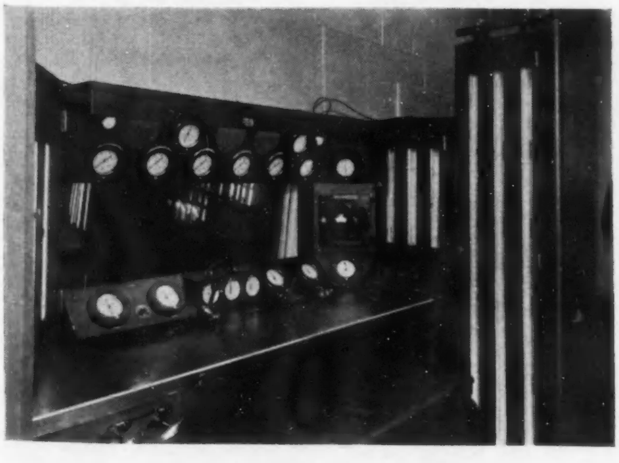
Various types of rotameters are available providing more rapid and safer means of measuring fuel flow and are sometimes more accurate than makeshift equipment used for calibration. Rotameters are satisfactory when fuel density is held reasonably constant and accurate calibration methods are used.

Oil consumption and flow are determined by change in weight of a tank mounted on a scale. Flow readings are determined by trapping scavenge oil in an overhead tank and checking the time required to flow a given quantity from the scale tank.

■ Instrumentation

Probably the greatest expression of personal tastes occurs in the design of the instrument panel and control desk. There is more latitude possible on these items without endangering operation than is true with other portions of the test cells.

The general motif for production testing where one man operates the engine, takes all data, and makes computations, is the console arrangement of instruments and controls.

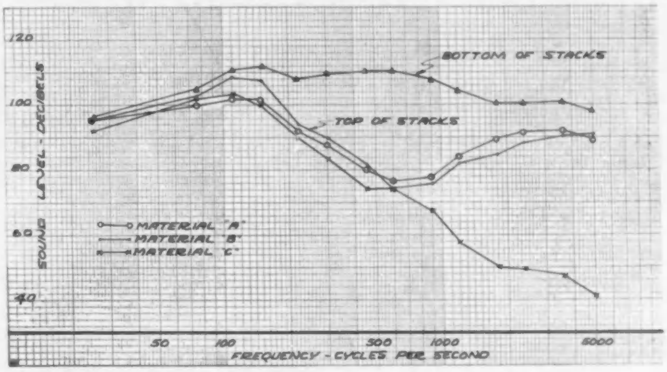


■ Fig. 10 – Control desk (new)

Fig. 9 shows the first control desk used in production. All controls, with exception of propeller pitch change, are through direct mechanical linkage.

All temperature and pressure indicating instruments are of aircraft type which permit close grouping in a small panel. The disadvantage, however, is the difficulty of reading accurately and easily instruments designed for indicating approximate conditions only.

Fig. 10 shows a later design of control desk incorporating precision instruments which include manually balanced potentiometer together with multipoint selector switch, mercury vapor thermometers, fluorescent illuminated manometers, hydraulic controls for throttle and mixture levers of carburetors, selector switches for temperature regulating valves, cable-driven tachometer counter-timer unit, and warning devices for low fuel and oil pressures.



■ Fig. 11 – Frequency distribution of noise reduction due to different sound-insulating materials

It will be noted that there is sufficient room for increased instrumentation expansion on the later design. This provision is always necessary, although not to the extent required in experimental testing.

■ Sound Treatment

It is absolutely imperative that test cells be equipped with sound-absorption treatment at least sufficiently so that operations do not become objectionable to nearby communities. The extent of treatment is determined to some degree on the location. The existing noise level must also be taken into consideration.

Three different materials have been studied for sound absorption under conditions of actual engine operations. Fig. 11 shows the noise level measured at the bottom of the treated stacks in the engine room and at the top of the stacks. It is readily apparent that there is little difference in absorption ability of the three materials below a frequency of 600 cps. For higher frequency it is apparent that one material is capable of greater absorption. The fact that no material tested is efficient below frequencies of 200 cps causes much difficulty from a low-frequency beat occurring when running a number of engines at approximately the same speeds. The analysis shown does not include the "beat." However, there is sufficient intensity of sound at low frequencies from each engine, that good absorption is nearly impossible.

The Role of Surface Chemistry and Profile

by J. T. BURWELL
Massachusetts Institute of Technology

BOTH the geometry and the chemistry of bearing surfaces have a marked effect on performance under conditions of boundary lubrication and the salient features of these factors are discussed in this paper.

It is pointed out that at least one other quantity in addition to the root-mean-square roughness should be specified in grading surfaces for lubrication performance. The presence of loose material or "fuzz" on all commercially finished surfaces is noted.

Data are presented to show that one function of addition agents in oils is to mitigate the bad effect of poor surface finish. The affinity of lubricants for metal surfaces is discussed, and methods for experimentally measuring this property are outlined, together with results.

It is shown that a high affinity or "wettability" is a necessary but not a sufficient condition for a good boundary lubricant. Its molecules should also have the proper structure and the lubricant should contain a surface-active addition agent in adequate concentrations.

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THE general subject of lubrication is usually divided into two types of operation, full-fluid or hydrodynamic lubrication and boundary lubrication. These two parts have many distinguishing characteristics with respect to operating characteristics, conditions under which they occur, and physical properties of the system upon which they depend. As regards the latter we can say in general that full-fluid lubrication depends almost entirely on the viscosity of the lubricant and very little on the properties of the lubricant-bearing surface interface while, under conditions of boundary lubrication, the reverse is true. Of course, both depend on other bulk properties such as hardness and thermal conductivity of the bearing metal, heat capacity of the oil, and so on, but in the present discussion we shall limit ourselves to the chemistry and geometry of the interface between the oil and metal surface. Continuing the contrast between the two types of lubricated operations, we note that, with knowledge of the viscosity alone, we are able to give a very complete description of hydrodynamic operation based solely on the laws of classical fluid mechanics. On the other hand, to obtain more than a most superficial picture of boundary-lubrication performance, it is necessary to invoke the atomic picture of the structure of matter and follow out all its consequences. In this paper we shall try to survey briefly whatever this picture can tell us about what goes on between boundary-lubricated surfaces.

The Bearing Surface

We may begin by first subdividing the subject into two parts and discussing in turn the contributions that the bearing surface and the lubricant each make to the performance of the bearing. The structure of a solid surface composed of even a commercially pure metal is, from the present point of view, a relatively complex affair. Each crystalline grain of the metal is made up of the metal atoms arranged on a relatively simple space lattice, to be sure, but these grains, in turn, are packed closely together without voids to form the bulk metal. This structure introduces complications. The transition region, a few atoms thick, between adjoining grains probably consists of atoms in a relatively disorganized state. Impurity atoms usually concentrate here. Physical properties of the grain interior such as plasticity and cohesive energy may show sharp changes at these grain boundaries with the result that, in the actual production of a piece of metal, high and nonuniform stresses may concentrate here. This is no less true of the grain boundaries which appear at the metal surface, and such stresses will result in a variation of chemical activity of the surface from point to point. This variation may affect the affinity of the lubricant for the surface, either by direct attraction of the oil itself or by the preferential attraction of gases or foreign impurities. Thus the adhesion of the oil may be either increased or decreased. This is particularly true in industrial practice where the surfaces, no matter how they are cleaned, are almost al-

[This paper was presented at the National Aeronautic Meeting of the Society, New York, N. Y., March 12, 1942.]

[1] See "Aggregation and Flow of Solids," by G. T. Beilby, Macmillan Co., New York, 1921.

[2] See *Naturwissenschaften*, Vol. 11, 1932, p. 112: Erg. exakt., by F. Kirchner.

[3] See *Physical Review*, Vol. 43, 1933, pp. 724-726: "Diffraction of Electrons by Metal Surfaces;" Vol. 49, 1936, pp. 163-166: "Diffuse Rings Produced by Electron Scattering;" both by L. H. Germer.

[4] See the *Journal of Chemical Physics*, Vol. 6, 1938, p. 749, by J. T. Burwell.

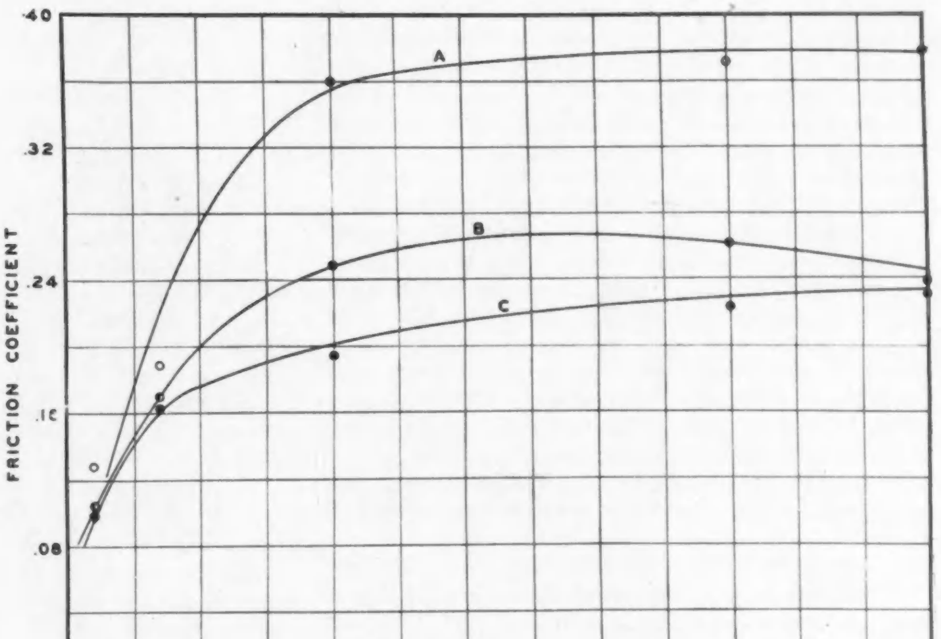
[5] See *Naturwissenschaften*, Vol. 16, 1937, p. 363, by G. I. Finch and H. Wilman.

in BOUNDARY LUBRICATION

ways contaminated. Another source of inhomogeneity of metal surfaces is that the different crystal faces of a grain may vary physically and chemically and, in polycrystalline metals having random crystal orientation, each grain will expose a different face to the surface. Certain mechanical working and recrystallization treatments may produce a preferential orientation of one or two faces in the surface, a technique which is utilized in the manufacture of commercial magnetically permeable steels. This is the state of affairs in an essentially one phase system. If, in addition, we have more than one phase such as SbSn and Cu_3Sn crystals in the tin-rich matrix in babbitt, or the lead in copper-lead alloys or the cementite in steel, then it is to be expected that the local variation of surface properties will be even greater. Worse yet is the adventitious presence of foreign materials such as slag inclusions or small fissures or voids.

In general, any surface shaping or finishing operation will contribute to the residual lattice strains in the metal by cold work. In cases, this may be so severe as to produce a metallurgical phase change and, in all cases, the grain size will be reduced. These grains may or may not acquire a preferred orientation. The more severe the operation, the smaller the grain size, and it was originally suggested by Beilby[1] that, after such an operation as polishing, the atoms near the surface are all in a completely disordered or amorphous state. The most direct evidence for this condition, such as is afforded by electron diffraction, is actually inconclusive[2, 3, 4], but the wealth of indirect evidence[1, 5] makes it seem very probable that the grain size in a polished or burnished metal surface is extremely small. Whether the truly amorphous state, which is the limiting case of a monatomic grain, is actually reached may be a question of degree rather than kind. However, such surface treatment will, in general, lessen the local variations in the surface properties by a process of smearing metal over the grain boundaries and submicroscopic cracks. On the other hand, where stresses may concentrate, such a surface may be chemically more active due to residual strain than is the bulk material. In this connection the interesting fact[5] has been discovered using electron diffraction that in the process of abrading cast iron the graphite is pulled out and smeared over the surface with the flat faces of the graphite particles adhering to the metal surface. These particles overlay the surface, preventing contact of the cast iron with the other rubbing surface. Here is an instance where direct evidence is afforded for the mechanism of the running-in of cast-iron surfaces.

So much for the picture of the surface on an atomic scale. In addition, one must consider a surface geometry of a larger order of magnitude, namely the surface roughness such as is produced by commercial finishing operations. These operations, besides cold-working the metal, reducing the grain size and sometimes affecting the metal-



■ Fig. 1 – Effect of roughness of ground and superfinished surfaces on friction coefficient using different lubricants – Curve *A*, pure mineral oil; Curve *B*, mineral oil plus 2% oleic acid; Curve *C*, pure oleic acid

lurgy as just mentioned, will impart to the surface its profile, and this is very important from the lubrication point of view. The subject of the geometry of a metal surface and methods of measuring it[6, 7] has been discussed quite completely elsewhere, and only a few points can be mentioned here. The coarser operations, such as turning and rough grinding, produce surfaces with the greatest roughness, that is, the height of the peaks above the valleys is the largest. These surfaces also have a steeper average slope, and this latter quantity is also important. This means in general a smaller true area of contact after the elastic deformation produced when two surfaces are brought into contact. This smaller true area of contact means higher unit pressures and hence severer conditions for the boundary lubricant whose function is to prevent metal-to-metal contact. For this reason the root-mean-square average slope is as important as the root-mean-square roughness itself in lubrication problems and should certainly be specified. It is taken care of in a qualitative way by specifying the type of finish, but this is not always as reproducible as one would like. The mechanical tracer type of profilometer at present on the market easily could measure both roughness and slope with little additional complication or expense.

The finer types of commercial finish, such as fine grinding, honing, superfinishing, and lapping, give smoother surfaces with an average slope that is surprisingly small, only a few degrees at most. It would appear on a magnified scale much like the contour of our middle-western plain. One would not expect this result from a visual observation of the very definite scratch pattern, but the truth is that the optical law that the angle of reflection must equal the angle of incidence is obeyed to within very narrow limits, and the few degrees difference in slope on two sides of a scratch can make one side appear light and the other dark, which is the means whereby we visually detect it.

In Table 1 are shown values of friction coefficient of surfaces of different roughness. These measurements were obtained on a pendulum type of machine similar to that described by Kyropoulos and Shobert[8] but with automatic recording of the decrement. The rubbing surfaces were crossed cylinders in the form of standard automobile piston pins on which surfaces of various roughness were produced. The roughness readings are root-mean-square values and were read with an Abbott profilometer. The SAE oil was a standard commercial lubricating oil made from a Pennsylvania crude and whose exact composition was not known. In Fig. 1 some of the results for ground surfaces of various roughness are plotted. The value for the superfinished surface of 2 micro-in. is included because it seems to fall in line with the values for the ground surfaces. The interesting fact appears that the friction increases with roughness up to about 20 micro-in. after which it remains essentially constant. The slight drop in some of the curves at 65 micro-in. may be due to the surface which these specimens had. To produce such a rough surface by grinding a very fast feed had to be used which imparted a slight spiral texture which undoubtedly increased the roughness as read by the profilometer. The scratches left by the individual grits were not nearly so rough and may have been really only 30 or 40 micro-in. or even less. That the type of finishing operation is important is shown by the results for the grit-blasted surfaces which are appreciably less than those for ground surfaces of the same roughness. This may be rather the fault of the roughness measurement and substantiates the fact that a second quantity beside the roughness is needed in comparing surfaces prepared in different ways. It may be further noted that, the smoother the surface, the less improving effect the addition agent has. Probably the incentive for producing improved addition agents in the oil industry would have been much less if our rubbing surfaces had always had good surface finishes. The lack of a good surface often can cause a great many headaches.

Table 1 – Effect of Surface Finish and Oil Addition Agent on Friction Coefficient

Surface	Superfinished	Ground	Ground	Ground	Ground	Grit-blasted
Roughness in micro-in. rms	2	7	20	50	65	55
Mineral oil	0.128	0.189	0.360	0.372	0.378	0.212
Mineral oil + 2% oleic acid	0.116	0.170	0.249	0.261	0.230	0.164
Oleic acid	0.099	0.163	0.195	0.222	0.238	0.195
Mineral oil + 2% sulfonated sperm oil	0.095	0.137	0.175	0.251	0.197	0.165
SAE 30 oil	0.119	.....	0.252	0.253	0.239	0.192

In addition to the regular hill-and-valley profile which such surfaces have, there also appear to be present in more or less amount some of the metal chips which were ploughed up in the formation of the valley but which either are not completely detached or are held to the solid by weak surface forces. The presence of such "fuzz," as the machinist would call it, seems to be characteristic of all abrading operations, but their size and quantity varies with the roughness of the finish. It is probably not present on finely polished or burnished surfaces. The removal of this loose material is one of the several changes that take place during running-in[9].

The anisotropy or directional quality of surface profile is another quality which distinguishes surface finishes. Turned and ground surfaces possess this quality, while honed, superfinished and grit-blasted surfaces, for instance, do not. This property can markedly affect bearing performance depending on the relation of scratch direction to the direction of motion.

It has been the purpose of this section to indicate some of the ways in which a metal surface, though made of a given material, can vary and hence affect its lubrication. This condition also points a moral for the investigator in this branch of lubrication – namely, that it is very difficult to produce uniform reproducible metal bearing surfaces for experimental study because of all the factors just mentioned. The greatest care should be exercised in all stages of their preparation with every variable taken into account unless it can be shown definitely that such a variable has no effect upon the results. It might be argued that such careful preparation of specimens is superfluous since surfaces in practice are never so carefully prepared. This is indeed true, but it must be borne in mind that there is only one way in which a surface can be clean while there are a thousand ways in which it may be dirty and it will never be dirty the same way twice. Any systematic study of the

[6]See "Technische Oberflächenkunde," by G. Schmaltz, J. Springer, Berlin, 1936.

[7]See 1940 MIT Conference on Friction and Surface Finish, p. 44, by S. Way.

[8]See *Review of Scientific Instruments*, Vol. 8, 1937, pp. 151-158: "A Simple Method of Measuring the Coefficient of Nonviscous Friction of Thin Lubricating Layers," by S. Kyropoulos and E. I. Shobert.

[9]See ASME Transactions, Vol. 63, 1941 (Journal of Applied Mechanics, Vol. 8), pp. A49-A59: "Effects of Surface Finish," by J. T. Burwell, J. Kaye, D. W. van Nymegen, and D. A. Morgan.

lubrication problem must begin with clean metal surfaces and pure lubricants, and then the study can be extended by taking into account the commonly occurring complicating factors one by one.

The Lubricant

Turning now to the structure of the lubricant, we can distinguish at least three properties that a good boundary lubricant should possess. (By boundary lubricant we shall mean here the conventional "oiliness" type of lubricant whose function is mainly physical, and we shall not discuss here the chemical polishing type of addition agent which has been investigated recently by Beeck[10] and his co-workers.) These properties are that the oil should have a high energy of adhesion, that the structure of its molecules be such that it can form a close-packed film of relatively large thickness, and that, if only a portion of its molecules be surface active, these be present in sufficient concentration to repair the surface film quickly enough to prevent prolonged metal-to-metal contact. Each of these requirements is a necessary one although none alone is a sufficient condition for a good boundary lubricant. We are specifically not considering here the other properties which a lubricant that is actually used in internal-combustion engines should have, namely stability, anti-corrosive characteristics and detergent properties[11]. These properties do not fall properly within the limits of surface chemistry and its bearing on boundary lubrication, although chemical attack touches on it.

Energy of Adhesion

This property might be called loosely the "wettability" of oil for the bearing surface. It is a function of both the surface and the oil, but appears to depend much more on the latter. It has the dimensions of energy per unit area which means the work necessary to pull a unit area of oil from the corresponding area of metal surface.

To begin, it must be borne in mind that all solids attract all liquids somewhat although we loosely say that water does not "wet" paraffin or that mercury does not "wet" glass. Even in these cases work must be done to remove the liquid from the solid. The icing of airplane wings is an energy-of-adhesion problem. Whether the water is liquid or solid, the last layer of water molecules next to the solid surface is strongly bound and probably has two-dimensional rigidity much like a solid. The physical chemist explains this attraction[12] in terms of the van der Waals forces that all molecules including the non-polar ones possess plus dipole attractive forces that polar molecules have. In the case of metal surfaces the attraction is increased by the minor image force in the metal surface. The most extreme attractive force may arise when an organic acid chemically attacks the metal surface slightly so that a layer of the metal ester is formed. Here the attracting force is ionic in type. The adhesion of oleic acid to copper, which it is known to attack, initially at least, is probably of this type. The physical picture is that the molecules of the lubricant, which are generally pictured as long and rod-shaped, stand on end on the surface. They generally have an active group or radical at one end, and this radical is strongly held to the surface by the aforementioned forces. The rest of the molecule is flexible and trails away from the active group. If isolated, it probably lies flat on the surface attracted there by weaker forces. As more and more molecules are crowded onto the surface, the active ends take up all the available space and the long chains are forced to stick up away from the surface much like the pile on a carpet. When there is lubricant in bulk covering the surface, this surface layer of molecules is a close-packed one, the next layer adhering to the first one in a less regular fashion and so on, so that, within a relatively few molecular distances from the surface, the bulk structure of the liquid is reached.

There are apparently only two ways of measuring the energy of adhesion[13]. One is by the heat of wetting and the other is by the contact angle. In the first method the heat liberated or absorbed when the liquid is added to the solid is measured and from this the energy of adhesion can be calculated, provided some assumption is made as regards the dependence of this energy on temperature. There are two objections to the application of this method to the present problem: First, one must use the solid in the form of a powder in order to provide enough surface to produce a measurable heat change and this is not the state in which it exists in the normal bearing surface. Secondly, it is the energy per unit area that is required and there is no good method for measuring the surface area of powders.

For this reason we turn to the second method which involves the measurement of the angle with which the surface of the liquid meets the solid. The energy of adhesion, W, is then calculated by means of the following equation:

$$W = \sigma (1 + \cos \theta) \tag{1}$$

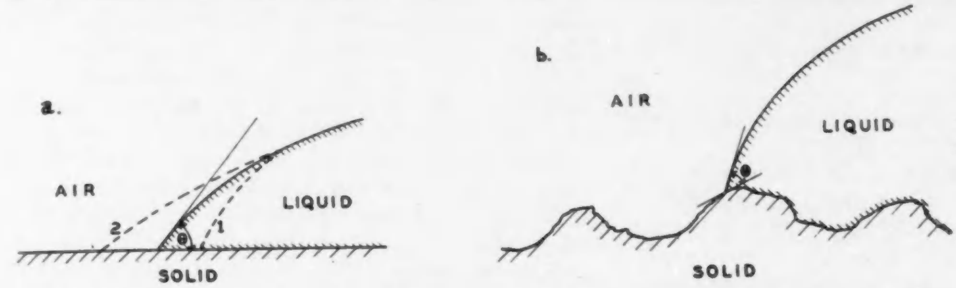
Here σ is the surface tension of the liquid and θ is the contact angle. We see from this that, for a given solid and liquid, W and σ are determined and so θ is determined. Hence, when a quantity of liquid is placed on a surface, it spreads until it reaches an equilibrium position when its shape is governed by its surface tension, its specific gravity, and by the angle at which its surface meets the solid. There is only one such equilibrium position and, if displaced either by constricting it or attempting to spread it, it will always return to approximately the equilibrium position. This phenomenon is familiar to anyone who has attempted to spread water on a slightly greasy surface. From this we can see how surface roughness can affect the spreading of oil on a bearing surface. The liquid in Fig. 2a will spread as long as its advancing surface makes an angle with the surface, such as in dotted position 1, greater than the equilibrium contact angle shown by the full line and, if artificially spread out, it will draw back as long as the receding surface makes an angle, such as in dotted position 2, less than the contact angle. Now, if the surface instead of being smooth has a profile such as is shown in Fig. 2b, then it is seen that the liquid may come to rest in the position shown regardless of where its position on the corresponding smooth surface might have been. There might be a lot of the liquid piled up behind the surface and gravity would tend to make it spread, but it will not advance down the slope because then the contact angle would be less than the equilibrium one. On the

[10]See Proceedings of the Royal Society, Vol. A177, 1940, pp. 103-118: "Mechanism of Boundary Lubrication – II. Wear Prevention of Addition Agents," by O. Beeck, J. W. Givens, and E. C. Williams.

[11]See SAE Transactions, March, 1941, pp. 98-106: "Lubricating Oils for Internal-Combustion Engines," by L. H. Mulit and F. W. Kavanagh.

[12]See p. 19: "Recent Advances in Surface Chemistry and Chemical Physics," by W. D. Harkins, The Science Press, 1939.

[13]See Chapter V: "The Physics of Chemistry of Surfaces," by N. K. Adam, Second Edition, Oxford University Press, 1938.



■ Fig. 2 – Adhesion of liquid to (*a*) a smooth surface, and (*b*) a rough surface

other hand, it might have been initially spread too thin so that the surface tension would want to pull it back, but it will not recede because then the contact angle would have to be less than the equilibrium one. The result is that a rough surface contributes to the hysteresis or spread between the advancing and receding angles of the liquid over the surface. If once spread, it will stay spread farther than on a smooth surface but, if initially added in one spot, it will not spread as far as it would otherwise. It would appear that, from this point of view, a roughness can only be detrimental for oils that have small contact angles (if the angle is actually zero it will spread over any surface) but, if for any reason one has to use an oil with a large contact angle, the rough surface should be advantageous. This may also explain the observation of engine manufacturers that oil sometimes spreads better on a roughened surface but it would probably be more proper to say that it stays spread better on a rough surface and does not as readily recede into drops. It is very doubtful whether an oil will spread better spontaneously on a rough surface than on a smooth one.

If, as may often be so, $W > 2\sigma$, then Equation (1) no longer holds since the cosine can never be less than one. In this case the liquid spreads until its contact angle is zero and then, since the surface forces are still not satisfied, it will continue to spread more or less indefinitely. The creeping of machine oils over surfaces is a familiar example of this condition, and indeed in the system metal surface-lubricating oil the contact angles seem to be always very small if not zero. In Table 2 the contact angles of a purified mineral oil which has the least affinity of all oils on various metal surfaces, is tabulated and the angles are all seen to be very small. (The significance of the difference between the advancing and receding angles will be discussed a little later.) It would appear then that we are balked in our attempt to measure the energy of adhesion by the method of contact angles. There is, however, another possibility. If we immerse the drop of liquid A to be studied on the metal surface in a second liquid B, then we can calculate the difference in energies of adhesion of the two liquids for the solid surface by the following equation[14]:

$$\Delta W = W_A - W_B = \sigma_A^* - \sigma_B^* + \sigma_{AB} \cos \theta_{AB} \qquad (2)$$

Here σ_A^* and σ_B^* are the surface tensions of liquids A and B respectively after equilibrium has been reached, σ_{AB} is their interfacial tension and θ_{AB} is the angle the liquid-liquid interface makes with the solid surface as measured in liquid A. If liquid B is so chosen that its contact angle with the solid surface when used alone is finite, then W_B can be calculated from the Equation (1) and, if θ_{AB} is finite, then ΔW can be calculated from Equation (2) and hence W_A can be computed. The surface tensions of the two liquids are written with asterisks to indicate that they are not the tensions of the liquids in the pure state but rather after they have been in contact with each other and any mutual solution and selective adsorption at the interface has taken place. If the tensions of the pure liquids were used in Equation (2) then ΔW would include the work done in such solution and adsorption as well as the difference in adhesion to the metal and so would give an entirely misleading result. Actually to measure σ_A^* and σ_B^* is difficult. One could take care of mutual solution by saturating a quantity of each with the other and measuring the surface tensions independently but, if one or both contain surface-active constituents which would concentrate at the liquid-liquid interface, we could never be sure of obtaining the same concentration of this active constituent at the surface when the tension of either liquid alone is measured. For most systems, however, with which we have to deal in this work under the conditions outlined above, Antonow's rule holds[15]. This rule states that the interfacial tension is equal to the difference in tensions of the two liquid surfaces *in the same state* as when they are in contact. Hence

$$\sigma_{AB} = \pm (\sigma_A - \sigma_B) \qquad (3)$$

If this is substituted into Equation (2), we obtain:

$$\Delta W = W_A - W_B = \sigma_{AB}(\pm 1 + \cos \theta_{AB}) \qquad (4)$$

The positive sign is used if liquid A in contact with B has a greater tension than B in contact with A. The negative sign is used if the reverse is true. This equation, like Equation (1), has the advantage that only one tension and an angle have to be measured.

There are several methods of measuring the contact angle but the one which seems to lend itself most appropriately to this type of work is the method of sessile drops. In this method a drop of the liquid is placed on the surface and allowed to spread to its equilibrium shape. Three dimensions are measured, either on the drop directly or on

Table 2 – Contact Angle of Purified Mineral Oil on Various Metal Surfaces

Metal	Contact Angle, deg Advancing	Receding
Silver	5	0
Copper	5	0
SAE 1040 steel	7	0
Tin-base babbitt	6	0
Aluminum	5	0
18-8 stainless	0	0
Mica	13	0
Soda lime glass	7	1

[14] See p. 113: "Fifth Colloid Symposium Monograph," 1927, by F. E. Bartell and H. J. Osterhof.

[15] See *Journal Chimie Physique*, Vol. 5, 1907, pp. 372-385: "Sur la Tension Superficielle a la Limite de Deux Couches," by G. N. Antonow.

its magnified image projected on a screen. Either a table can be made up relating the ratio of two of these quantities empirically to a drop-shape parameter, such as has been done with pendant drops[16], which with the third measured dimension determines the contact angle, or the tables of Bashforth and Adams[17] may be used directly to determine the surface tension and contact angle. The latter method is more fundamentally sound but more laborious. The method applies whether the drop is in air or immersed in a second liquid. One advantage of this method is that simultaneous measurements can be made of both the tension and the angle which are required in Equations (1) and (4). A second advantage is that, due to the surface inhomogeneities discussed earlier, the contact angle will vary within rather wide limits over any metal surface no matter how carefully it is prepared, but the single measurement on the drop gives automatically an average of the angle all around the periphery of the circle of contact between drop and metal. The method has the disadvantage of being somewhat time-consuming. A more complete description of the method and results obtained with it will be published elsewhere.

As pointed out before, great care must be exercised in preparing the specimens. It is now generally agreed that the only way to obtain a fresh uncontaminated metal surface is by abrasion[18]. No solvent treatment is satisfactory. Washing with soap leaves a monolayer of sodium stearate on rinsing. Abrasion on emery papers is not entirely satisfactory as it has been found that the glue or binder on the paper may be picked up. Also the loose abrasive may charge up the surfaces of the softer metals. The most satisfactory procedure yet found has been to abrade on an Arkansas stone or a porcelain streak plate. The abrasion should be carried out under a liquid to prevent local surface heating and oxidation, preferably benzene or a light petroleum fraction provided these can be obtained and maintained free of greasy contaminant; otherwise under de-aerated water.

Table 3 – Contact Angle and Adhesion Energy of Acetylene Tetrabromide for Solid Surfaces

Surface	Contact Angle, deg Advancing	Contact Angle, deg Receding	Adhesion Energy
Mica	25	20	93.5 ergs/cm^2
Soda lime glass	32	24	91.5 ergs/cm^2

Some results are shown in Tables 2, 3, and 4. These show a characteristic common to all contact angle measurements, namely that the angle measured when the liquid is advancing over the surface is always greater than when receding from it[13]. In other words there appears to be some hysteresis in the system. Various reasons have been suggested for this result. Roughness of the surface could produce it as discussed previously, but the present surfaces were too smooth for that factor to have much effect. There may be passive resistances which retard the motion of system toward equilibrium; in other words, the two angles might approach the same value if enough time is allowed. Again it has been suggested that the two systems are actually different and hence should have different adhesion energies. In the advancing case the solid has not yet been touched by the liquid but may be covered with an adsorbed gas film or some contamination. This film the liquid may remove but on recession may leave an adsorbed film of its own. Certain it is that, the cleaner the surface, the less the hysteresis. This fact again points to the relatively large effect of mere traces of impurity and the great need of careful preparation of surfaces. In actual lubrication work both angles, and the energies calculated therefrom, may have practical significance. The advancing angle measures the ability of the lubricant to spread into the clearance between moving parts and to re-cover regions of the bearing surface that have been wiped or scraped clean. The receding angle measures the work necessary to wipe the surface free of lubricant. The receding angle should certainly be zero for all lubricants else a tangential stress on the bulk lubricant will remove it from the surface ($W < 2\sigma$).

Table 4 – Difference Between Adhesion Energies of Various Oils and That of Water for 18-8 Stainless Steel

Lubricant	$W_{oil} - W_{H_2O}$, ergs/cm^2
Pure mineral oil	−83.2
Pure mineral oil + 2% oleic acid	− 3.2
Pure mineral oil + 2% stearic acid	− 3.1
Pure mineral oil + 1% cetyl alcohol	− 1.1
Pure mineral oil + 2% hexyl chloride	− 6.7
Pure mineral oil + 2% methyl stearate	− 5.6
Pure oleic acid	− 2.3
Commercial SAE 30 lubricating oil	−33.1

In Table 2 the advancing angle appears to be small but finite. A consideration of Equation (1) shows that W need be only very slightly larger for θ to be zero since the cosine θ changes so slowly in this region, and since the receding angle was zero in most cases the finite advancing angle probably has not much significance. Hence we can say that the contact angle of pure mineral oil for most surfaces is zero and Equation (1) cannot be applied. Oils with addition agents will have even greater affinity for the metal. We must look for another liquid which makes a finite angle with the metal so that Equation (4) may be used. So far the search has been unsuccessful although a great variety of liquids have been tried. Some of these liquids would make finite angles on non-metal surfaces, as shown in Table 3. In these cases adhesion energies can be calculated. It might even be argued that since the surface tensions of a wide variety of liquids as measured by the ring tensiometer, which must assume zero contact angle with the platinum ring, agrees so well with those measured by independent methods, then all these liquids must completely wet platinum and probably likewise the other common metals. Mercury is no good as a reference liquid because of its tendency to amalgamate and also because of the susceptibility of its surface tension to contamination[19].

Table 4, however, shows the sort of measurements that can be made. For a given metal we can compare the relative affinities of different lubricants by calculating their

[16]See *Journal of Physical Chemistry*, Vol. 42, 1938, pp. 1001-1019: "Boundary Tension by Pendant Drops," by J. M. Andreas, E. A. Hauser, and W. B. Tucker.
[17]See "An Attempt To Test the Theories of Capillary Action," by F. B. Bashforth and J. C. Adams.
[18]See Philosophical Transactions of the Royal Society, London, Vol. A234, 1935, p. 329, by W. G. Beare and F. P. Bowden.
[19]See "Surface Tension and the Spreading of Liquids," by R. S. Burdon, Cambridge University Press, 1940.

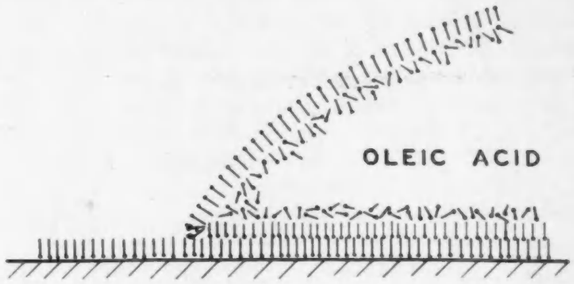
Table 5 – Effect of Concentration of Oleic Acid in Mineral Oil on Friction Coefficient of Ground Surfaces of 20 Micro-in.

Lubricant	Friction Coefficient
Pure mineral oil	0.360
2% oleic acid in mineral oil	0.249
10% oleic acid in mineral oil	0.198
50% oleic acid in mineral oil	0.198
Pure oleic acid	0.195

ΔW's by Equation (4) with reference to the adhesion energy of some immiscible liquid such as water for the metal. This latter energy is unknown but constant for all the cases. When it is borne in mind that this value must be more than twice the surface tension of water or in excess of 150 ergs/cm^2, then it is seen that all the addition agents to pure mineral oil raise its energy of adhesion to about the same value independently of whether the active radical was hydroxyl, carboxyl or halogen. The energy for all these mixtures was about the same as that of pure oleic acid. It is known that the first fraction of a percent of addition agent will completely cover the solid surface with a monomolecular film, and this is sufficient to change the energy to that typical of the pure addition agent.

An interesting phenomenon was observed which elucidates some of these mechanisms in terms of the molecular structure of the lubricant. It was observed that oleic acid would not spread on copper and, if spread out, it would recede to an angle of about 25 deg. This behavior was substantiated on the other metals and also on glass and mica. This result was unexpected because of its known efficacy as an oiliness addition agent in reducing friction (see Table 5). Further investigation showed that water would not wet regions of a surface where the oleic acid had been and, further, that the oleic acid would not spread on a glass plate that had been wiped with mineral oil nor would it spread on paraffin. These facts give a clue to the reason for this behavior. Oleic acid consists of a straight hydrocarbon chain of 15 carbon atoms with a carboxyl group on one end. As the liquid spreads a monolayer of molecules is tightly adsorbed on the metal surface with the carboxyl groups next to the metal as shown in Fig. 3, and the long hydrocarbon chains packed closely together and sticking more or less normally upward. The advancing free surface of the oleic acid drop also has an oriented surface layer of molecules with the carboxyl ends pointing out. The affinity of such carboxyl groups, which are hydrophilic, for a hydrocarbon surface, be it made of paraffin or an oleic acid monolayer, is small and surface tension holds the advancing surface back. With a 2% solution of oleic acid in mineral oil the liquid drop again deposits an oleic acid monolayer on the solid, but now the advancing free surface of the drop is, due to thermal agitation in the liquid, composed statistically of a large number of hydrocarbon molecules which have a great affinity for a hydrocarbon surface. Such a solution will spread without difficulty on a metal. Lauric alcohol and cetyl alcohol behave in a similar manner. With the latter the action is so marked that its advancing angle over a truly clean surface is less than its receding angle from the regions where it has been, which is the reverse of what is ordinarily observed. It must not be concluded from this, however, that these compounds do not have a great affinity for metal surfaces. On the contrary, the first monolayer adheres quite strongly and is difficult to remove, but the bulk liquid is then reluctant to spread over the monolayer. This then gives a good reason for using small percentages of these addition agents in oil if one wants bulk spreading of the lubricant. There are, of course, other reasons for not using 100% oiliness addition agent such as drying, expense, and less wide range of viscosity.

The small affinity of a liquid for a solid or the immiscibility of two liquids generally can be explained in terms of molecular structure. Organic liquids whose molecules are predominantly composed of hydrocarbon groups, whether aromatic or aliphatic, are quite soluble in each other and quite immiscible in liquids whose structures are composed largely of hydroxyl or carboxyl groups such as



■ Fig. 3 – The non-spreading of oleic acid on its own monolayer adsorbed on a solid surface

water, methyl alcohol, acetic acid, glycerol, and so on. These latter liquids, on the other hand, are completely miscible in each other.

It has been pointed out that high adhesion energy is partly responsible for the great efficacy of graphite in oil lubrication. The graphite particles stick to the metal and it has been found that most oils have a great affinity for graphite[20], probably accounted for by the carbon-carbon linkage which is common to both structures.

■ Molecular Structure of the Lubricant

We turn now to the molecular structure of the lubricant and its importance to any boundary lubricant. The high energy of adhesion, discussed in the preceding section, is due almost entirely to the active radical, such as hydroxyl, carboxyl, halogen, amine, and so on, and practically not at all to the bulky hydrocarbon portion of the molecule. On this basis acetic acid and stearic acid would be expected to have about the same energy of adhesion for a metal, and yet their lubricating efficiencies are widely different. Acetic acid has only one carbon atom linked to the carboxyl group and its close packed monolayer will be very thin, whereas in stearic acid a chain of 15 carbon atoms are linked to the carboxyl group and its monolayer is roughly 15 times as thick. The latter is also much better as a boundary lubricant. Hence high energy of adhesion is not the only quality necessary for a good boundary lubrication. The molecule must in addition have the correct shape. A long straight chain seems to be best with the active or adhesive radical on one end. Molecules which are flat and plate-like

[20] See p. 225: "The Physics and Chemistry of Surfaces," by N. K. Adam, Second Edition, Oxford University Press, 1938.

or which have aromatic rings incorporated in their structure either lie flat on the surface thus giving a relatively thin monolayer or, if they do stand up, they cannot pack as closely together in the plane of the surface. With long chains one can obtain a lateral close packing of the chains and an equally close packing of the active radicals which means a maximum adhesion per unit area of film[21].

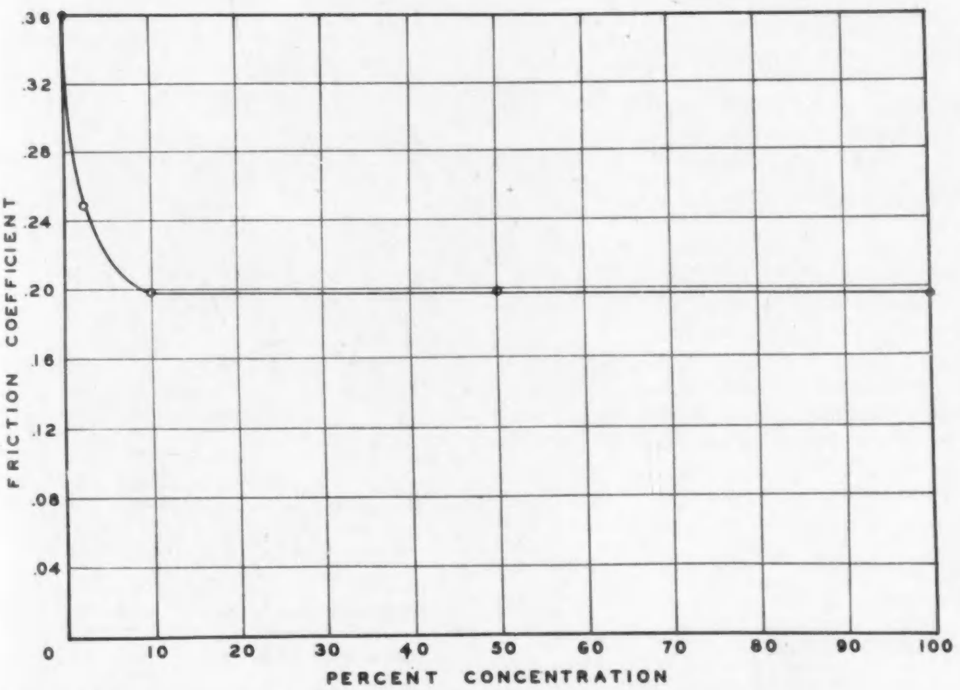
Using homologous series of straight carbon chain organic compounds with different active radicals, Sir William Hardy[22] and his co-workers found a very definite dependence of friction coefficient not only on the active radical but also for a given radical on the length of the hydrocarbon chain. The friction coefficient decreased as the chain length increased, an equal decrement for each CH_2 group added to length. This decrement was independent of the solid surface used.

■ Concentration of Surface-Active Material

Study of the data in Table 5 which is plotted in Fig. 4 gives an indication of the third prerequisite of a good boundary lubricant. Here is tabulated the friction coefficient of various concentrations of oleic acid in mineral oil on ground surfaces of 20 micro-in. Table 4 shows that the adhesion energy of all the solutions containing any oleic acid is closely the same. The molecular shape of the oleic acid and the mineral oil is about the same, namely a straight chain of about 16 carbon atoms. Nevertheless, we see that the friction coefficient decreases quite appreciably as the concentration of oleic acid is increased up to about 10%, whereas the energy of adhesion does not change at all after the first trace of oleic acid is added. This latter fact is because the first fraction of a per cent of oleic acid added all selectively adsorbs at the oil-metal interface and is sufficient to cover it completely. The reason for the efficacy of increased concentration from a friction point of view is clear. When one surface is rubbed over another, separated by only a thin film of lubricant, which may be more or less tightly bound to each surface, the high points of each rub and the local pressure is generally enough to rupture or tear away the lubricant film. If this film is not repaired quickly the same local area of bare metal continues to rub against the other surface, heat is produced, the temperature rises and local welding and seizure may occur[23]. This is all on a very small scale of dimensions but each such local weld contributes to raise the friction. If, on the other hand, the concentration of surface-active material in the lubricant is great enough, these molecules can diffuse to the surface and re-coat it quickly, so quickly that, before the next high spot on the opposing surface can rub over the area in question, the latter has become covered with lubricant and no bare metal is exposed.

The question may be asked: in Table 5 it appears that 100% oleic acid gives the lowest friction coefficient and yet it was stated earlier that oleic acid would not spread. How then is it such a good lubricant? It is true that it would not spread on its own monolayer but the latter itself is present and very tightly bound to the metal. If the oleic acid is present in sufficient quantity to flood the bearing, it will repair the monolayer but, if only present in small drops on one part of the surface, it will not reach and repair the ruptured monolayer on another part. Here is a case where a rough surface, by preventing the oleic acid from receding into drops, would be advantageous.

[21]See p. 226: "The Physics and Chemistry of Surfaces," by N. K. Adam, Second Edition, Oxford University Press, 1938.
[22]See p. 227: "The Physics and Chemistry of Surfaces," by N. K. Adam, Second Edition, Oxford University Press, 1938.
[23]See Proceedings of the Royal Society, Vol. A169, 1939, pp. 371-391: "The Nature of Sliding and Analysis of Friction," by F. P. Bowden and L. Leben.



■ Fig. 4 – Effect of concentration of oleic acid in mineral oil on friction coefficient

■ Conclusions and Acknowledgments

We have tried to show very briefly in this paper in what ways the geometry of the bearing surfaces and the chemistry of the oil-metal interface may affect boundary-lubricated operation. In the main this discussion has made use of the molecular picture of the structure of matter and has served not only to elucidate some otherwise unexplained phenomena but also to indicate by inference the lines that future investigation should take.

The author is indebted to the Chrysler Corp. for the funds which have supported the research on surface finish and lubrication in this laboratory for the past three years, to W. Mikelson of the General Electric Co. for preparing the grit-blasted specimens, and to J. M. Mochel and H. W. Fox for assistance in developing the method for measuring the adhesion energies and in taking data.

1941 CFR ROAD DETONATION TESTS

The cooperative road tests carried out during 1941 have added considerable information and experience to that already existing on the subject of road detonation testing.

Extensive data were obtained on the fuel requirements of the 1940 and 1941 models of the three most popular cars. Corresponding data were obtained on the knocking characteristics of current gasolines representing the bulk of the sales volume in various parts of the United States.

On account of large variations in octane-number requirement among different cars of the same make – due to differences in ignition timing, combustion-chamber deposit, and other causes – and on account of variations in commercial gasolines, it has been necessary to use statistical methods of analysis in the appraisal of fuel and engine relationships.

These methods of analysis have been applied in a number of ways, and have proved very useful. For this reason, the continuance of cooperative activity in compiling current statistical information annually on fuels and car requirements is recommended.

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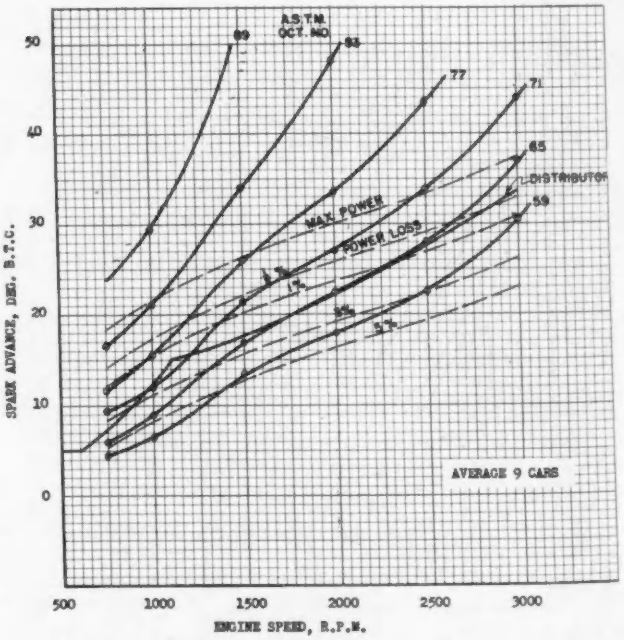
THE AUTHORS: For 16 years prior to his appointment as secretary of the Cooperative Research Council, C. B. VEAL (M '12) was SAE research manager, and secretary of the CFR Committee. During and after World War I he was with Curtiss Aeroplane & Motor Co. in a consulting capacity. At one time he was associated in consulting practice with the late C. M. Manly. Mr. Veal taught at Purdue for some years after his graduation from that University. JOHN M. CAMPBELL (M '37), who has presented many papers before the SAE, has concentrated his work at GM Laboratories on detonation, methods of eliminating it, and ways of measuring its magnitude. In this connection he was active in the CFR tests at San Bernardino in 1940. WILLIAM M. HOLADAY (M '28) has been closely identified with development work in both fuels and lubricants. He was a member of the original Uniontown Road Test Group in 1932 and has participated in many of the subsequent field activities of the SAE. He is a research engineer in the general laboratories of the Socony-Vacuum Oil Co. R. J. GREENSHIELDS, whose specialty is the testing of motor fuels, has been brought into close contact with the work of the CFR Committee, particularly the tests carried out at Uniontown, Pa., and San Bernardino, Calif. The use of borderline spark advance as a method of evaluating motor fuels, which he proposed, has been accepted by the CFR as the approved method of conducting road tests. He graduated from the University of Illinois in 1932 with a B. S. degree.

■ Program

DURING 1941 the Cooperative Fuel Research Committee continued its investigation of the knocking characteristics of fuels and engines in actual service. The centralized road knock tests, made at San Bernardino more than a year ago,[5] resulted in the development of various road-test procedures. At the same time the tests supplied data illustrative of certain new principles, which appeared to have useful application as a means of expressing some of the engineering relationships between fuels and engines. The data obtained at that time, however, although they were valuable and suggestive, were not considered sufficiently extensive to warrant unqualified recommendation of these procedures and methods of analyses without further experience with them in the hands of individual laboratories. Accordingly, the program carried out during the past year has been designed to test these procedures and methods of



■ Fig. 1 – Borderline reference-fuel framework on 1941 car "A" with power loss for various degrees of spark retard superimposed

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 16, 1942.]

[1]Research Laboratories Division, General Motors Corp.
[2]Shell Oil Co., Inc.
[3]Socony-Vacuum Oil Co., Inc.
[4]Secretary, Cooperative Research Council and CFR Committee.

[5]See SAE Transactions, May, 1941, pp. 193-204; API Proceedings, Vol. 22M, Section III, Tulsa, Okla., May 19-22, 1941, pp. 119-134: "1940 Road Detonation Tests," by J. M. Campbell, R. J. Greenshields, and W. M. Holaday.

– Further Experience with New Methods

(Compiled from Report of the Cooperative Fuel Research Committee)

by J. M. CAMPBELL,[1] R. J. GREENSHIELDS,[2] W. M. HOLADAY,[3] and C. B. VEAL[4]

analysis, and to obtain more definite information with regard to their limitations as indicated by more extensive data.

The 1941 road detonation program was divided into sections, as follows:

1. Ignition-timing relationships.
2. Control fuel tests.
3. Procedure evaluation.
4. Survey of characteristics of commercial gasolines.
5. Survey of octane-number requirements of 1940 and 1941 cars, with particular attention to Chevrolet, Ford, and Plymouth models.

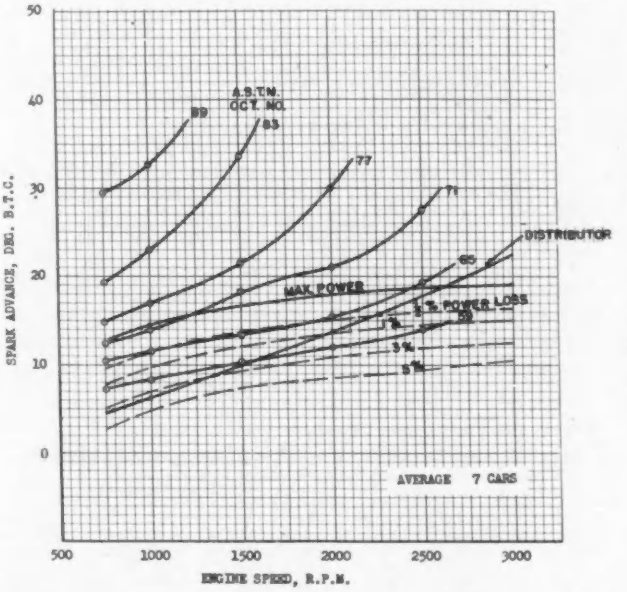
The last section, relating to octane-number requirements, was carried out in conjunction with the program of the Survey Division of the Cooperative Fuel Research Committee, and will not be considered in detail in this report.

■ Ignition-Timing Relationships

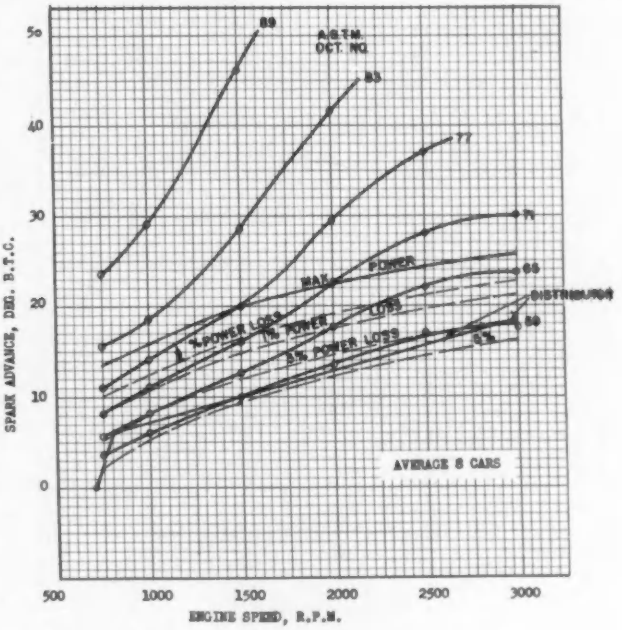
In any given engine, ignition timing has a controlling influence on the knocking tendency, or octane-number requirement. At the same time, ignition timing has an influence on power. Accordingly, when deciding on the most desirable ignition advance throughout the speed range, both power and detonation must be considered – usually resulting in a compromise between the retardation required for knock control and the advance necessary for power.

The so-called reference-fuel framework,[5] upon which ignition timing for maximum power and fractions thereof are superimposed, are a convenient means of expressing these ignition-timing relationships at wide-open throttle. Reference-fuel frameworks were developed for 1940 and 1941 models of Chevrolet, Ford, and Plymouth. Although space does not permit the inclusion of all of these data, representative average frameworks for the 1941 models are shown in Figs. 1, 2, and 3.

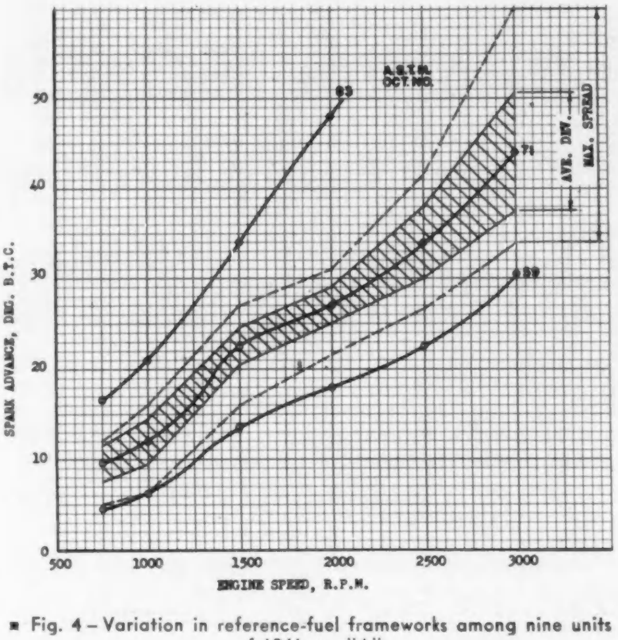
Figs. 1, 2, and 3 show the signal importance of ignition timing in controlling knock. For example, retarding the ignition 5 deg from the setting required for maximum power at 1000 rpm (approximately 20 mph) resulted in a loss of about 1% in power and a reduction in the octane-number requirement of from 6 to 10 octane numbers in these cars. The position of the curve representing the manufacturer's specification for distributor advance indicates that each of the three manufacturers has taken ad-



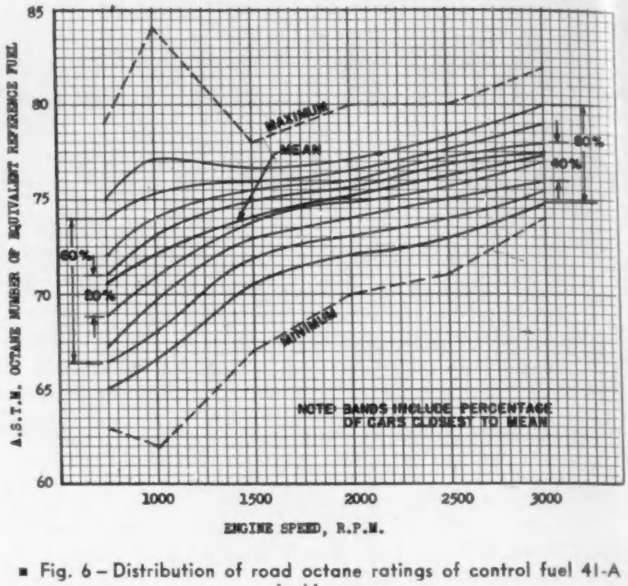
■ Fig. 2 – Borderline reference-fuel framework on 1941 car "B" with power loss for various degrees of spark retard superimposed



■ Fig. 3 – Borderline reference-fuel framework on 1941 car "C" with power loss for various degrees of spark retard superimposed



■ Fig. 4 – Variation in reference-fuel frameworks among nine units of 1941 car "A"

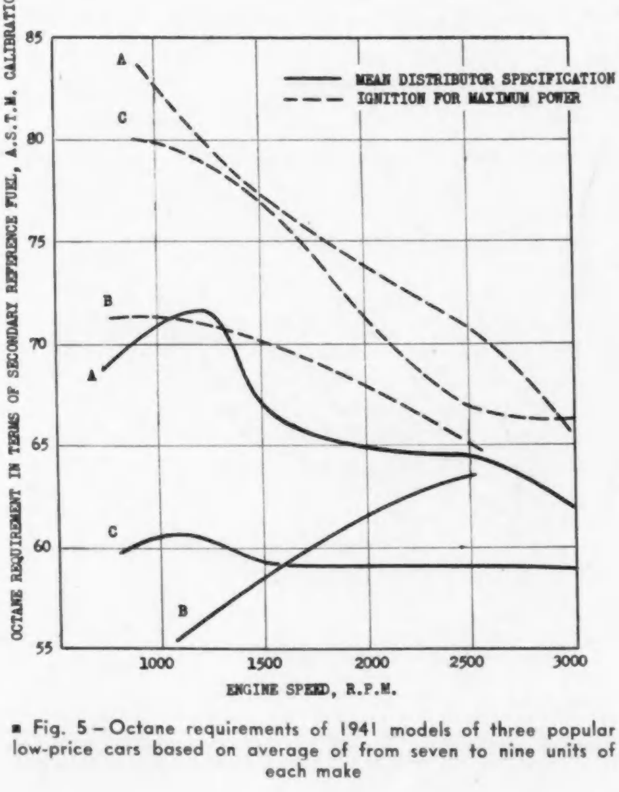


■ Fig. 6 – Distribution of road octane ratings of control fuel 41-A in 66 cars

vantage of this principle by retarding the ignition relative to the setting for maximum power at the lower engine speeds. The position of the several different distributor curves represents the individual manufacturer's compromise of a number of factors, including power loss from retarded spark, octane-number of available fuels, allowance for the effects of combustion-chamber deposit, allowance for deviations from proper ignition setting, and other elements peculiar to the design of each engine.

The most important limitation on the value of these reference-fuel frameworks is the variation from car to car of the same make. For example, Fig. 4 illustrates the variations reported among nine different units of one make of car. This figure is based on the average value for each car, and the use of individual rather than average values for each car, of course, would result in an even greater spread. At high speed, the spread in the average deviation was about 12 deg, and at low speed about 4 deg. At either speed, this is the equivalent of from 8 to 10 octane-numbers.

This spread presumably is due to a number of causes, such as real differences between engines, errors due to inaccuracies in instrumentation for observing spark advance, and other sources of experimental error. Whatever the causes, these variations between cars have important significance in that they indicate the necessity for the use of statistical methods in the evaluation of the octane-number requirements of any particular make of car, as found in customer service. The application of such statistical methods in "octane-number requirement" surveys will be discussed further in a subsequent part of this report.



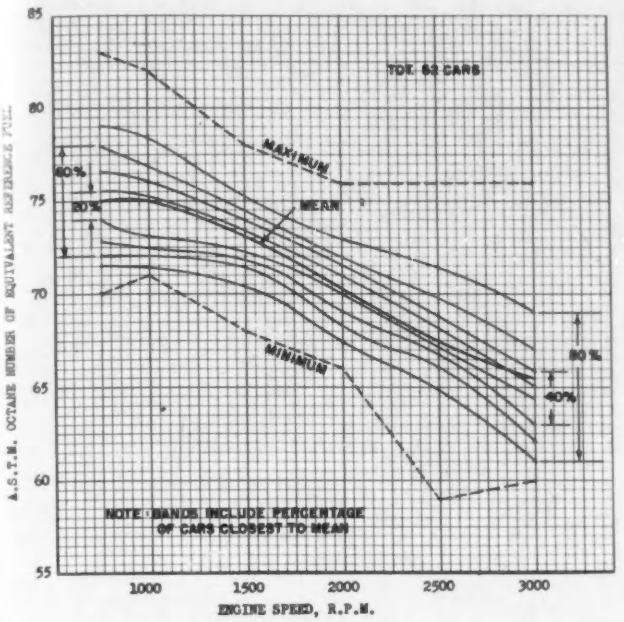
■ Fig. 5 – Octane requirements of 1941 models of three popular low-price cars based on average of from seven to nine units of each make

■ Requirements from Frameworks

The reference-fuel frameworks shown in Figs. 1, 2, and 3, inasmuch as they are based on average data from a number of different units of each make, probably are the most representative evaluations available for each make of car. It is, therefore, of some interest to determine the average octane-number requirement of each of these three makes, which represent more than 50% of total production for 1941. These requirements are shown in Fig. 5.

Table 1 - Average Inspection Data on Four Control Fuels Used in 1941 Road Tests

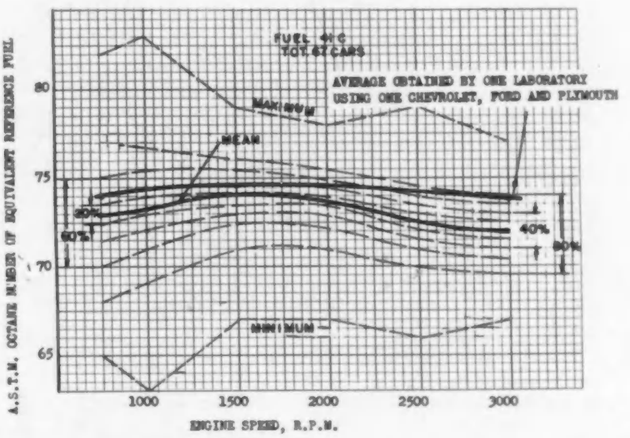
Code Letter	ASTM Octane No.	Research Octane No.	ASTM Distillation, F: Initial Boiling Point	10% Point	50% Point	90% Point	End Point
41-A	75.1	75.5	109	166	245	328	374
41-B	69.4	79.7	93	126	247	370	414
41-C	75.6	79.7	98	132	224	323	379
41-D	75.9	78.6	117	159	199	248	335



■ Fig. 7 – Distribution of road octane ratings of control fuel 41-B in 52 cars

It must be emphasized that the ratings in Table 2 are average values. The individual data show considerable variation in octane-number at each engine speed. This variation appears due in part to differences between cars, and in part to other factors which might be classified as experimental error. The differences between various makes of cars – that is, Chevrolet, Ford, and Plymouth – were not very consistent, however, and present information is not sufficient to indicate real and unmistakable differences between these different car makes. But, for whatever reason, variations in reported octane-number ratings do exist; and it is necessary, therefore, to obtain a statistical representation of the octane-number speed relationship as it was actually reported for these tests. Such a representation is given in Figs. 6, 7, 8, and 9. In these figures the distribution of more than 60 car ratings about the mean is shown. For example, Fig. 8 shows that 40% of the ratings reported on 41-C at 1500 rpm were between 73.0 and 74.7 octane-number.

Under these circumstances it is obvious that erroneous conclusions can be reached by using a limited amount of data. It appears further that control fuels such as were used in these tests may have a useful place in future road-test work. As an example of the use of a control fuel to check the consistency of data obtained by an individual laboratory, the average road octane-number data obtained by one laboratory on one Chevrolet, one Ford, and one



■ Fig. 8 – Distribution of road octane ratings of control fuel 41-C in 67 cars

■ Control Fuels

Four control fuels of widely different types were used by all of the cooperating laboratories in the 1941 road tests. A sufficient number of ratings was obtained to establish representative mean values of octane-number for each fuel throughout the speed range. Such data make it possible to investigate the variations in ratings from car to car of the same make, and to compare ratings in different makes of car. As one of the control fuels is a reference-fuel blend, certain deductions relative to experimental error can be made.

The control fuels were chosen to represent the following four different types of gasoline:

Paraffinic straight-run, containing tetraethyl lead.

Mid Continent cracked (no lead).

Mixture Mid Continent cracked, plus straight-run, and containing lead.

Secondary reference-fuel blend.

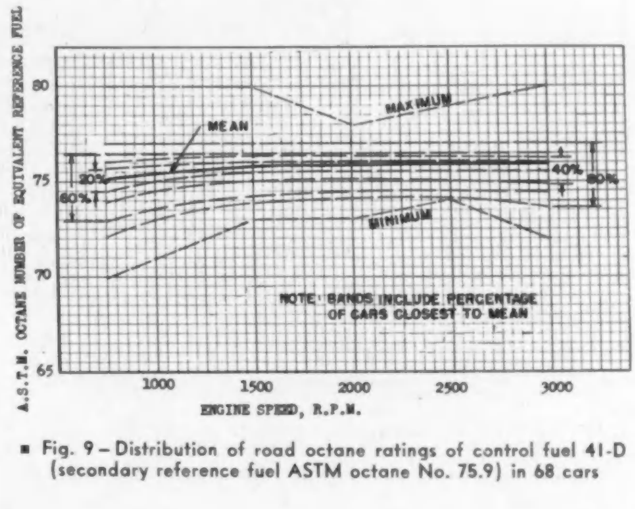
Average laboratory inspection data on these fuels are given in Table 1.

■ Octane Number Vs. Speed

One of the outstanding developments of the San Bernardino road detonation tests was the concept of a relationship between engine speed and road octane-number that can be used to characterize the behavior in service of different types of gasoline. This relationship is illustrated by the average road octane numbers, tabulated in Table 2, of the four control fuels used in the 1941 road tests.

Table 2 – Average Road Ratings of 1941 Control Fuels Expressed as ASTM Octane No. of Equivalent Secondary Reference Fuel

Code Letter	Engine Speed, rpm					
	750	1000	1500	2000	2500	3000
41-A	70.7	72.0	74.1	75.2	76.3	77.9
41-B	75.0	75.0	73.0	70.3	67.5	65.4
41-C	73.0	73.2	74.3	73.7	72.6	72.3
41-D	75.1	75.4	75.8	75.8	75.8	75.9



■ Fig. 9 – Distribution of road octane ratings of control fuel 41-D (secondary reference fuel ASTM octane No. 75.9) in 68 cars

Plymouth are shown in Fig. 8. In this case the average for 3 cars, including 1 of each make, was within about 1 octane number of the grand average at speeds below 2000 rpm. At higher speeds this divergence of the single laboratory average from the grand average increased to 1.7 octane numbers.

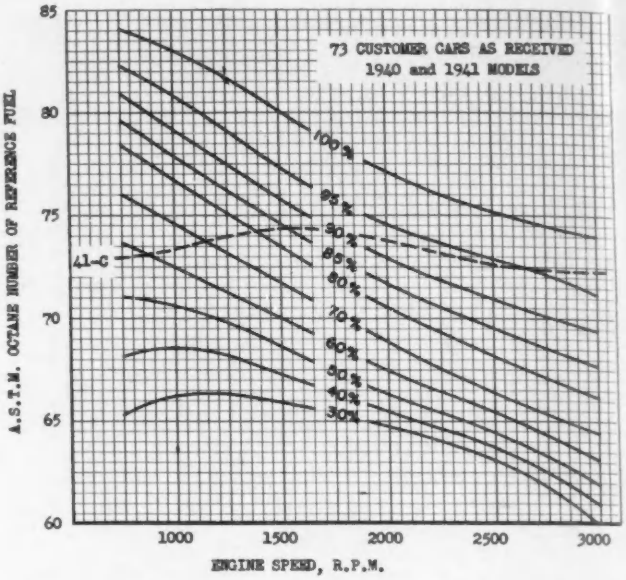
The conclusion to be reached from these data is that single ratings cannot be relied upon, owing to appreciable variations between runs in a given car and between different cars. At the same time, it cannot be emphasized too strongly that every precaution should be taken in obtaining the original road-test data to minimize the possibility of error in individual ratings.

■ Procedure Evaluation

Evaluation of the three road-test procedures[5] developed at San Bernardino for obtaining road octane values involves two considerations: (1) comparison of ratings obtained by different methods on a given fuel; and (2) comparative reproducibility of each method.

Fig. 10 represents an analysis of the three methods on this basis, using fuel 41-C as an example. In general, it appears that the three methods give approximately identical ratings for a given fuel. The differences found between different methods were of the order of one octane number – scarcely enough to be significant. The evaluation of any conclusion is conditioned further by the limited data available.

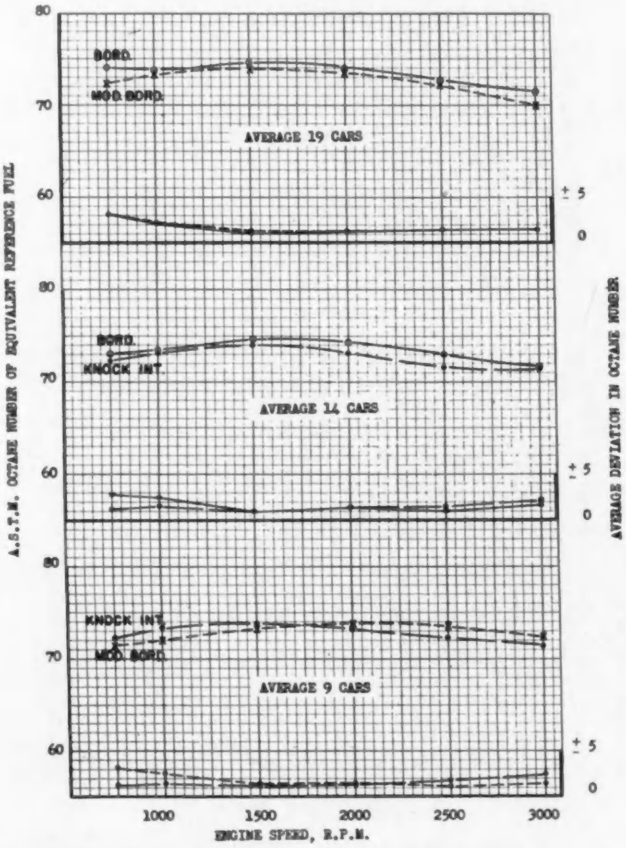
Reproducibility is expressed in Fig. 10 by the average-deviation curves, which indicate little or no difference between the borderline and modified-borderline methods in



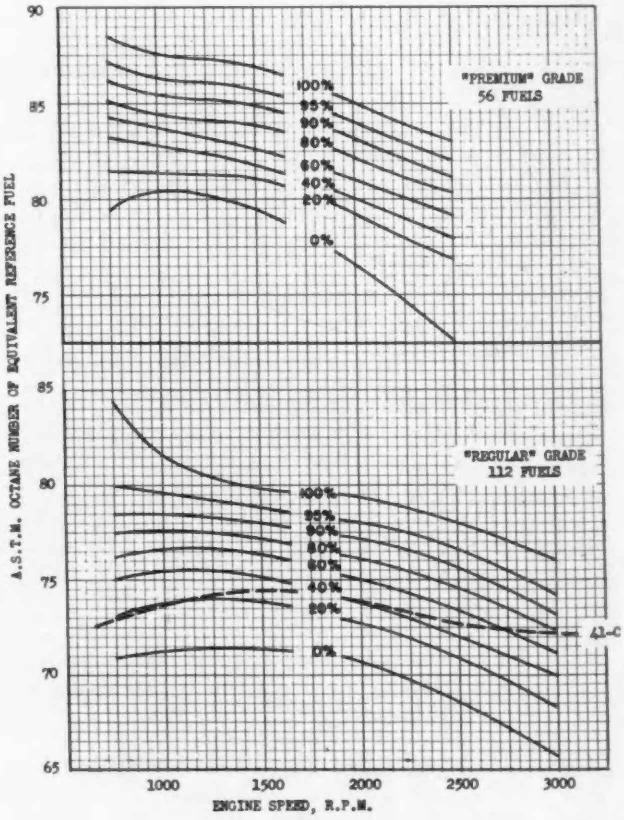
■ Fig. 12 – Percentage of cars giving borderline or no knock on reference-fuel blends

this respect. At low engine speeds the knock-intensity method appears to have an advantage, and this was shown in the rating of all 4 control fuels below 1000 rpm.

In general, it appears that the three procedures may be used interchangeably for obtaining road ratings. The choice of method depends somewhat upon its adaptability to the existing circumstances, and somewhat upon the individual preference of the operator. All three methods



■ Fig. 10 – Comparison of testing methods using control fuel 41-C



■ Fig. 11 – Distribution of road octane ratings of commercial fuels in all cars

require a considerable amount of skill on the part of the observers.

Commercial Gasoline Survey

To provide some statistics on the service characteristics of current commercial gasolines, each participating organization was asked to determine the octane number-speed characteristics of at least five "regular"-grade and two "premium"-grade gasolines. These fuels were to be selected to represent the major quantity of gasoline marketed in the area where each participant was located. By this means it was possible to obtain a reasonably representative sampling of gasolines marketed throughout the United States.

Data were reported on a total of 112 "regular" and 56 "premium" fuels, and the distribution of road octane-number ratings in all cars is shown in Fig. 11. The curves representing various fractions of the total were determined from a tabulation of octane number ratings at six different

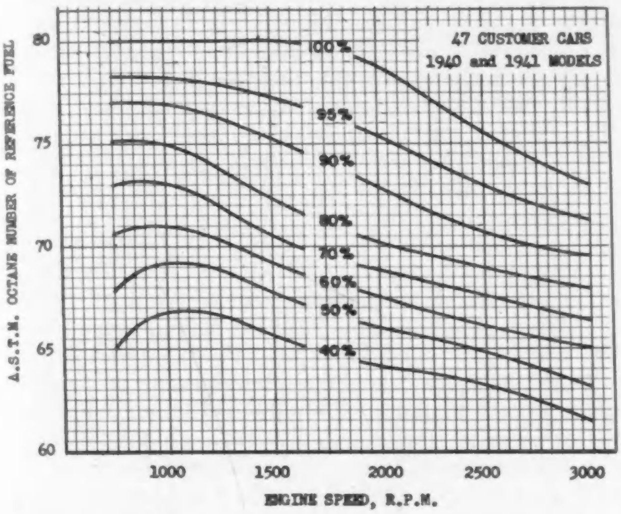


Fig. 13 – Percentage of cars giving borderline or no knock on reference-fuel blends after adjustment of initial spark timing in accordance with manufacturer's specification

speeds, and it should be pointed out that these curves bear no relation to the characteristics of the individual gasolines, that is, the gasolines in the upper 10% bracket at 1000 rpm are not necessarily the same as those in that bracket at 3000 rpm.

This type of information has several applications. For example, it can be used in the evaluation of the characteristics of any given fuel, such as 41-C, the curve for which is superimposed on the statistical framework representing the "regular"-grade gasolines. Thus 41-C was a little below the average at 750 rpm, and a little above the average at 3000 rpm.

By combining these statistics with corresponding octane-number requirement data for a given make of car, such as those shown in Fig. 5, the probabilities of detonation in various speed ranges may be estimated. This information may be used also in conjunction with reference-fuel frameworks as a guide in the design of distributor characteristics that will conform with fuels currently available. When they are available, of course, corresponding data for a particular car make would be preferable for distributor design, and the complete CFR report contains separate figures representing the behavior of "regular" and "premium" gasolines in 1940 and 1941 models of Chevrolet, Ford, and Plymouth.

Octane Requirement Survey

In a preceding section of this report reference was made to a statistical method of expressing octane-number requirements in a given make of car. For the information of the petroleum refiner, such an expression inclusive of several car makes would be even more desirable. To illustrate this type of information, Fig. 12 was prepared from the octane number requirement data obtained on 73 customer cars, as found in service, without adjustment. These cars included 24 Fords, 15 Chevrolets, 13 Plymouths, 7 Buicks, 5 Chryslers, and 9 cars of other make. Of these cars, 47 were tested again after adjustment of the initial spark timing to the setting recommended by the manufacturer, and the results of this second survey are shown in Fig. 13.

This type of information can be used as a guide in the analysis of gasoline characteristics in relation to car requirements. For example, superimposing the average road rating of gasoline 41-C on the data in Fig. 12 visually presents an estimate of the percentage of cars that can be operated on this fuel without knock. Thus at 750 rpm less than 50%, and at 3000 rpm less than 5%, of the cars will knock on this fuel. A similar analysis can be made for each car make, provided sufficient data are available.

Combining the fuel-survey data illustrated in Fig. 11 with octane-number requirement survey data of the type illustrated in Fig. 12 graphically presents the statistical relationship, for freedom from knock, between current fuel characteristics and car requirements. This relationship is shown in Fig. 14. Appraised by this comparison, the average "regular"-grade fuel will be satisfactory – operate without knock – in more than 75% of these cars at 1000 rpm (approximately 20 mph), and will be equally satisfactory in more than 90% of the cars at 3000 rpm (approximately 60 mph).

The validity of such conclusions depends, of course,

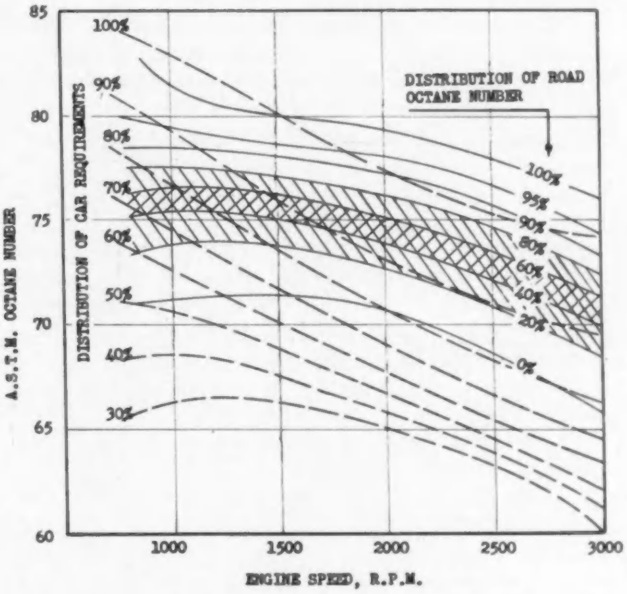


Fig. 14 – Relationship between road octane number of commercial "regular" grade gasolines and car requirements shown in Fig. 12

upon the availability of sufficiently representative data; and, for this purpose, the pooling of information from different laboratories is very desirable.

■ Acknowledgment

The CFR committee is indebted to the 23 organizations that contributed data to this project, and to the Analysis Group,[6] the members of which have given generously of their time and energy to the task of tabulating and analyzing the very considerable quantity of data involved.

■ Participating Organizations – 1941 Tests

The Atlantic Refining Co., Chrysler Corp., Cities Service Oil Co., Ethyl Gasoline Corp., General Motors Corp., General Petroleum Corp., Gulf Research and Development Co., Holley Carburetor Co., National Bureau of Standards, Phillips Petroleum Co., The Pure Oil Co., Shell Oil Co., Inc., Sinclair Refining Co., Socony-Vacuum Oil Co., Inc., Standard Oil Co. of Calif., Standard Oil Co. (Ind.), The Standard Oil Co. (Ohio), Standard Oil Development Co., Sun Oil Co., The Texas Co., Tide Water Associated Oil Co. (Bayonne), Tide Water Associated Oil Co. (California),[7] Union Oil Co., and Universal Oil Products Co.

[6] The personnel of the Analysis Group was: R. J. Greenshields (leader), Shell Oil Co., Inc.; M. J. Anderson, Ethyl Gasoline Corp.; H. W. Best, Yale University; K. Boldt, The Pure Oil Co.; W. E. Drinkard, Chrysler Corp.; J. G. Moxey, Jr., Sun Oil Co.; C. H. Van Hartesveldt, The Atlantic Refining Co.; C. B. Veal, Society of Automotive Engineers (now secretary, Cooperative Research Council); and T. A. Weir, Socony-Vacuum Oil Co., Inc.

[7] No data submitted.

Automobile Engineering Organization and Procedure

(*Concluded from page* 443)

by its organization chart. Cooperation is the factor which implements the organization chart and, in the last analysis, permits the organization plan really to function. Strict military observance of the chart is the exception rather than the rule, except where policy matters are involved, in which latter case it is followed rather closely.

3. Design and experimental functions cannot be entirely divorced. Here again, cooperation is one of the keynotes but, in addition, the importance of securing close supervision of design detail by experimental personnel is clearly established. The men charged with design development during laboratory, proving grounds, or road test, contribute heavily to the final result and can be of major assistance to the designer in the initial stages.

4. Under normal conditions of *successive continuing programs,* the horizontal-type organization has major advantages as compared with the vertical. These advantages increase with the size of the organization and the scope of operations. These advantages include:

a. Ability to accommodate personnel specialization.

b. More uniform work division.

c. Less organization dependence on individuals whose loss would be excessively felt.

d. Greater organization flexibility to accommodate variations in department loads.

5. The vertical-type organization has advantages where suitable experienced personnel are available, in small organizations, or where projects in general are not of the successive continuing type. Military developments of many types fall into this non-continuing classification as contrasted with the car yearly model programs.

6. As horizontal organizations increase in size, need arises for superposition of additional follow-up personnel.

7. The separate internal research function set up within the organization deriving its inspiration, ideas, and a certain amount of guidance from the organization as a whole, guided by a "research council," seems to have definite advantages in the automobile industry as compared with the external research group which tends, without extremely capable guidance, to drift too far from practice.

8. Control of engineering costs varies from a *minimum* for the large producer, who is primarily concerned with *product – cost per production unit* to a *maximum* control of engineering costs for the smaller producer who is required to consider *engineering cost per production unit.*

9. It seems to be agreed that an adequate system of test records is an economy, even though the organization is small and its staff overloaded. In the large organizations an extensive system of circulating written test reports is essential to keeping a large group informed. In the smaller organization, frequent staff meetings serve this latter purpose.

10. The *committee method* of reaching decisions on major design questions has wide acceptance. It provides mechanism for quickly reaching the most available facts. Usually men who face the same facts will draw substantially the same conclusions. The committee method removes uncertainty and substitutes confidence *in* and understanding *of* decisions reached. Men who understand *why* they seek an objective reach for it with zest and attain it much sooner. *Ability is actually increased by understanding why.*

11. Most organizations retain some follow-up on projects by means of dollar or time allocations. The positive use of dual dollar and time budgets for this purpose seems highly advantageous as a simple control procedure. Periodic reports of dual budget status issued by the follow-up office to project assignees maintains a balanced program effort.

12. Under normal competitive market conditions advantage must be taken, in the initial as well as final design phase, of accurate production cost estimating as a service function available to the designing and development groups. Major advantage accrues if this service is centered in the engineering organization. This arrangement brings production and design experience together while designs are still fluid and readily susceptible to modification. After designs have once crystallized, it is often too late to justify cost-saving changes due to effect on other affected parts.

13. Extension of laboratory facilities as a means of developing and testing new designs, preceding road tests, is finding greater approval as an economy measure. Laboratory facilities save valuable time and reduce the need of more costly road-test procedures. There is agreement that some road testing must be continued.

14. There is necessity for production-engineering facilities operating between engineering department and factory for the handling of alteration procedure, also associated with the inspection function, to permit combined detection and correction of faults. Closer association of these three functions than now provided seems advantageous.

15. Engineering becomes increasingly involved with production problems; production increasingly involved with engineering problems. A full combining of these two major functions might be advantageous.

Design Features of the JUNKERS 211B AIRCRAFT ENGINE

by SIDNEY OLDBERG and THOMAS M. BALL
Chrysler Corp.

THE Junkers 211B engine follows the usual German practice of very large displacements and conservative mean effective pressures and rotative speeds. However, the relative light weight per unit of displacement results in a net weight per horsepower that is not far above its competitors.

Fully automatic devices which control propeller speed, manifold pressure, mixture ratio, spark advance, and supercharger gear ratio follow the German policy of removing all possible distractions from the pilot.

This is one of three large liquid-cooled engines known to be produced in quantity in Germany; it powers an impressive percentage of the *Luftwaffe*. While of external appearance and displacement that resemble the Daimler-Benz DB-601 engine, the fundamental construction, detail design practice, and metallurgy of the Junkers 211B are surprisingly different.

With a few notable exceptions the materials analyzed betrayed little shortage of alloying elements, contents of these constituents averaging close to American aircraft practice.

In the main part of this paper, the authors conduct readers on a critical tour of the vitals of this complicated engine.

■ ■ ■

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THE Junkers 211B engine is one of the three large liquid-cooled aircraft engines known to be produced in quantity in Germany. It powers the Junkers JU-88 and Heinkel He 111K twin-engine bombers, the Junkers JU-87B single-engine dive bomber, and the Focke-Wulf FW 200K long-range four-engine bomber, altogether an impressive percentage of the *Luftwaffe*.

Its makers, the Junkers Flugzeug-Und-Motorenwerke, A.G., of Dessau, have produced the well-known Junkers diesel aero engine since 1929, and in 1937 introduced two gasoline engines of 1202 and 2136 cu in. displacement, the Jumo 210 and 211 series. In 1938, the carburetor of the 211 series was replaced by direct cylinder injection.

The engine examined, which was removed from a JU-88, had apparently run but a few hours since its last overhaul when captured, which circumstance permitted an observation of finish and fitting practice. Its cylinder block was poured in October, 1939, so that it does not represent the latest development, although cursory examination of an engine, the crankcase of which was poured more recently, showed no observable changes.

■ Appraisal

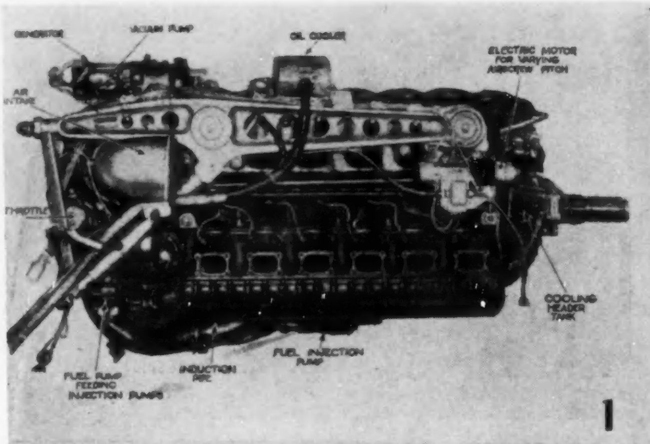
At the outset, several general impressions and observations should be pointed out which may lead to a better understanding of the discussion to follow.

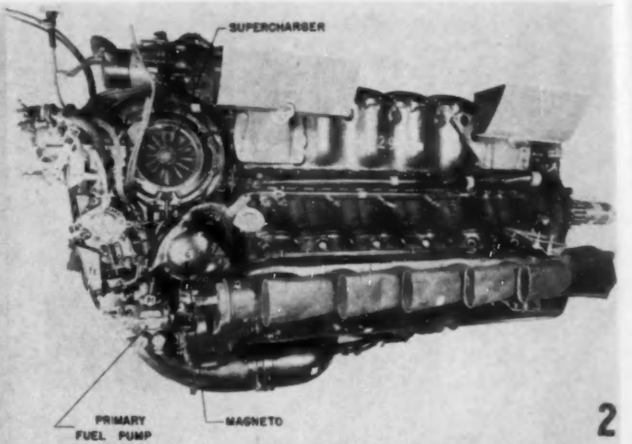
1. While of external appearance and displacement that resemble the Daimler-Benz 601 series[1], in that it is a 60-deg inverted-vee with direct gasoline injection, the fundamental construction, detail design practice, and metallurgy of the Junkers 211B are surprisingly different. No parts appear to be common to the two engines excepting, possibly, the starter and generator.

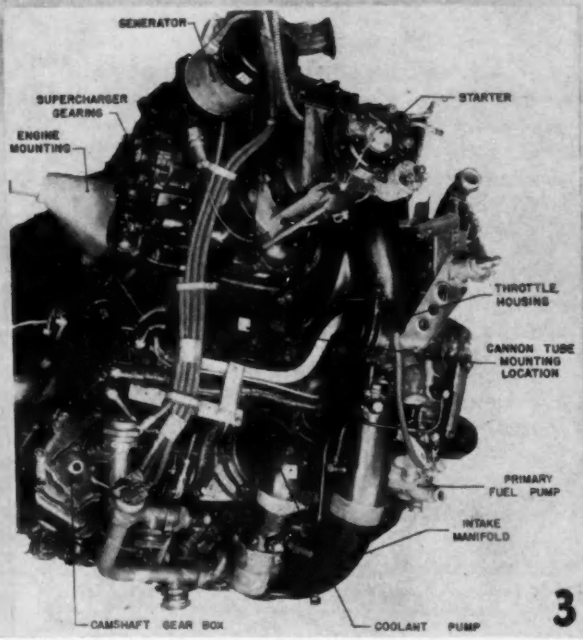
2. The general arrangement is logical and simple, although the manufacturing complications of several parts indicate an enviable ascendancy of the designer over the

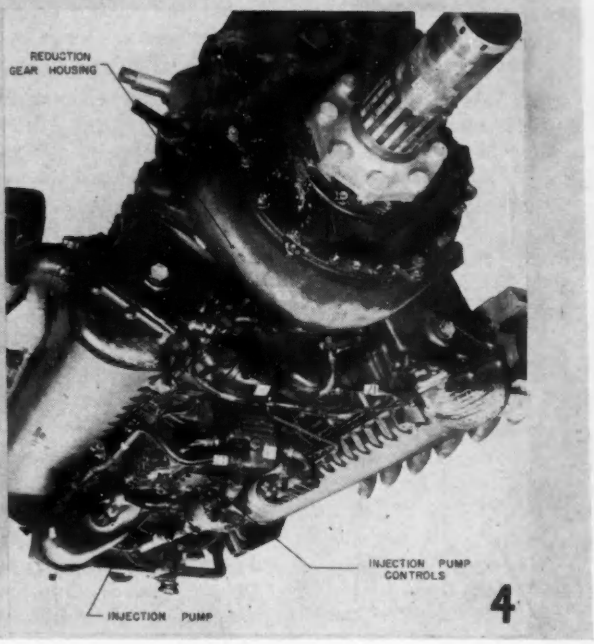
[This paper was presented at the National Aeronautic Meeting of the Society, New York, N. Y., March 12, 1942.]

[1] See SAE Transactions, October, 1941, pp. 409-431: "Mercedes-Benz DB-601A Aircraft Engine (Design Features and Performance Characteristics)," by Raymond W. Young.









■ Fig. 1 – Side view of Junkers 211B engine showing method of mounting

■ Fig. 2 – Engine – starboard side

■ Fig. 3 – Engine – rear

■ Fig. 4 – Engine – bottom front

production department. The solution of typical design problems for optimum stress distribution is carried to the most minute details.

3. In general, the engine was manufactured with care and precision. Parts expected to require service replacement appear to be interchangeable, although the machining errors in certain large castings were found to have been corrected by hand working in the mating pieces. Surface finish and dimensional uniformity of like parts equal American practice, with but few exceptions.

4. The materials analyzed betrayed little shortage of strategic alloying elements, contents of these constituents averaging close to American aircraft practice. In general, the steels used were drawn to hardnesses far below their usable upper limits, and the control of grain size was somewhat neglected. Whether soft to expedite machining or to decrease notch sensitivity, the physical properties observed in many of the steels could have been obtained more economically.

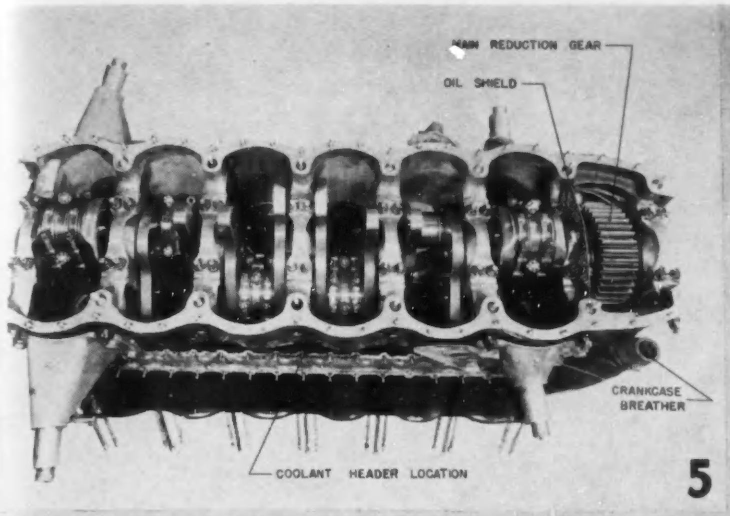
5. As shown by Appendix I, the specifications of the engine and a comparison with its contemporaries, the Junkers 211B follows the usual German practice of very large displacements and conservative mean effective pressures and rotative speeds. However, the relatively light weight per unit of displacement results in a net power output per pound that is not far below its competitors.

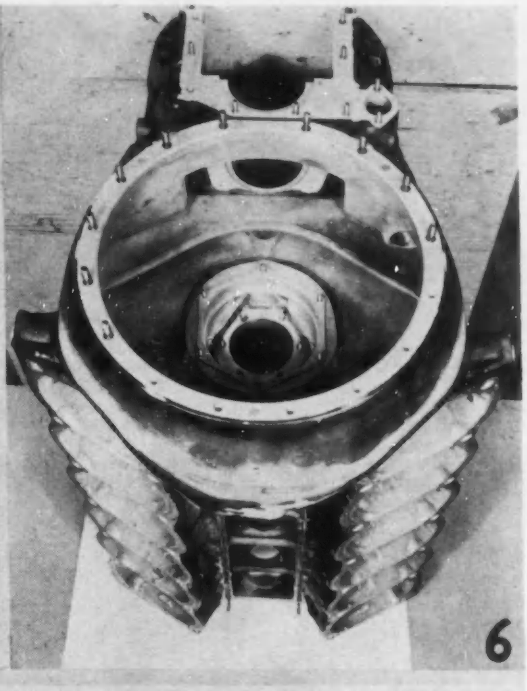
6. Fully automatic devices which controlled propeller speed, manifold pressure, mixture ratio, spark advance, and supercharger gear ratio follow the German policy of removing all possible distractions from the pilot.

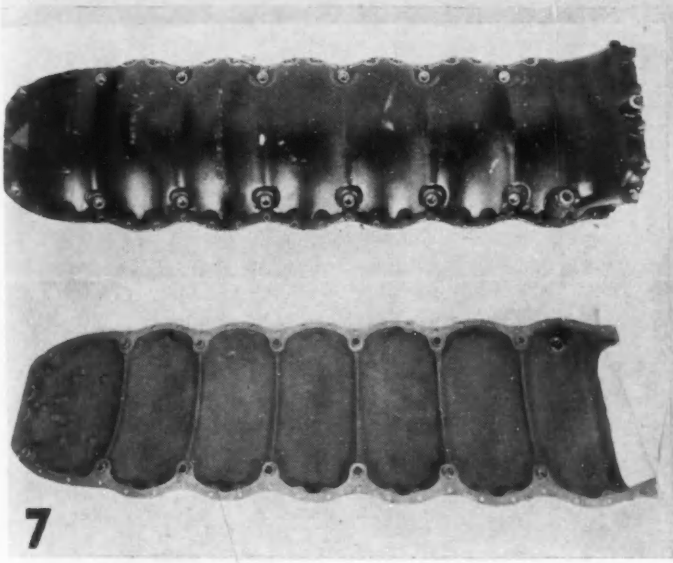
■ Installation

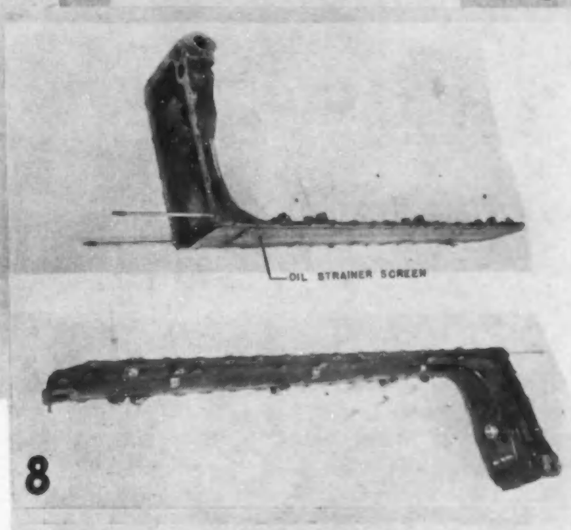
The known installations of this engine are bulkhead mounted, as in Fig. 1. The front section of the nacelle, the oil cooler and a horse-shoe-shaped coolant header were supported on the engine. In some installations[2] the radiator is directly below the engine, although from the parts examined the method of attachment was not clear. Line connections to the pilot's compartment were mostly jointed at the firewall, and all lines were color-coded.

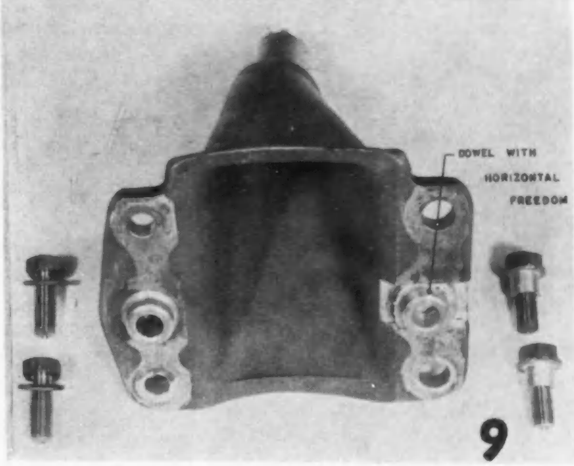
[2] See *Flight*, Vol. 37, No. 1621, Jan. 18, 1940, pp. 45-50: "The Junkers Petrol Injection Engine."











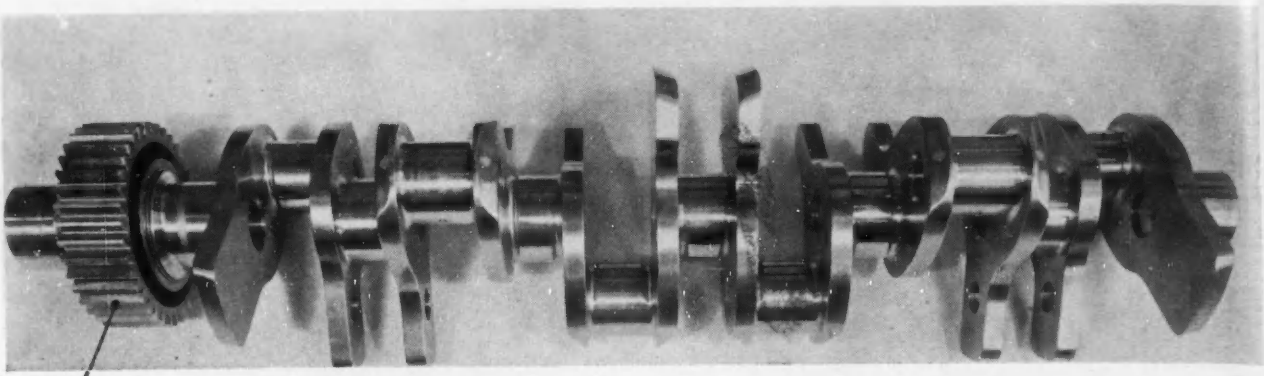
- Fig. 5 – Engine – top view showing crankcase
- Fig. 6 – Crankcase – front view
- Fig. 7 – Top crankcase coverplate
- Fig. 8 – Bottom crankcase coverplate
- Fig. 9 – Engine mounting

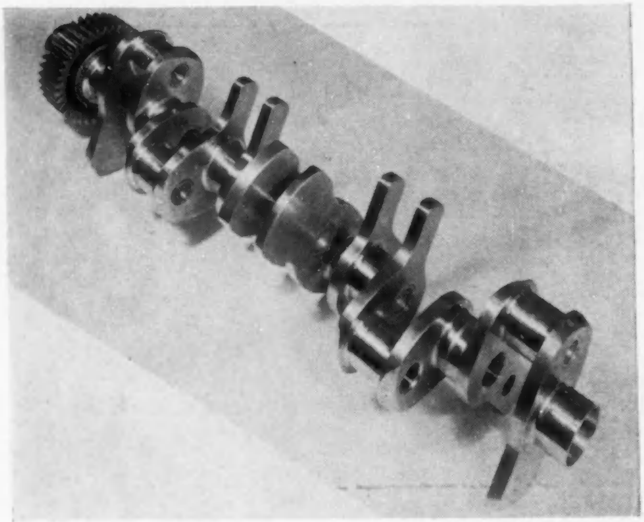
Constructional Details

The general arrangement is shown in Figs. 2, 3, and 4, which are more or less self-explanatory. Notable are the compact accessories section, the high mounting location, and the amount of external plumbing, particularly in the vee. Lengthwise, the engine is compact, having 6.62-in. cylinder centers with a 5.9-in. bore.

Crankcase – The crankcase is shown in Figs. 5 and 6. This part is of cast aluminum, of very substantial monoblock construction. Only the stressed ribs are hand-polished. The main bearing caps were very tightly fastened by hand scraping to a high interference fit within the case, and were further secured by cross bolts. The legs of the vee are tied together by ribs through which the cannon tube is run.

The reduction-gear support is shown in Fig. 6. An extra rib is seen which serves to baffle the gear throw-off from the breathers. The crankcase top cover, Fig. 7, is plainly a structural member, being well ribbed and staked with 14 hollow dowels to the crankcase. The dowel holes were evidently hand-reamed in place. The crankcase lower cover, Fig. 8, is not a structural member, but serves to collect the oil from the crankcase scavenging holes.





▪ Fig. 10 (above and at left) – Crankshaft

Mounting trunnions, Fig. 9, are of forged aluminum and secured to the crankcase by dowels, of which one has horizontal freedom in the flange to allow relative expansion between flange and case.

Crankshaft – The Junkers Jumo 211B crankshaft, Fig. 10, is notable in several respects:

1. Counterbalancing is carried out with the specific purpose of relieving main bearing loads, and the angular distribution of counterweights is optimum for this purpose. In addition to the crankcheeks, approximately 30 to 56% of the crankpin and connecting-rod rotating weight is counterbalanced.

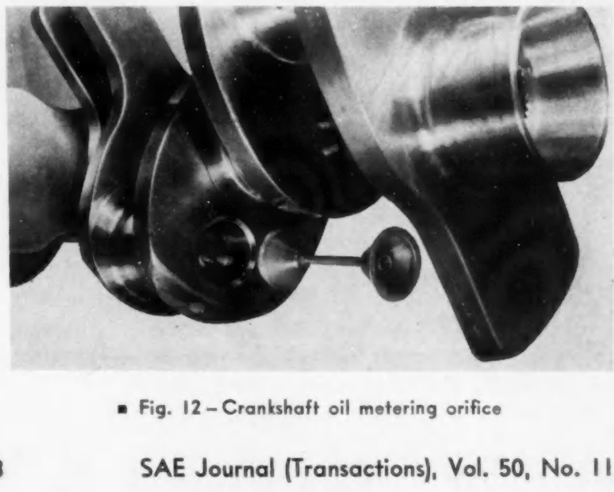
2. Large compound re-entrant fillets are used on the underneath semicircumference of each crankpin and on the semicircumference of each main journal adjoining the working part of the associated crankcheeks. In this way, a maximum fillet is obtained without reduction of bearing area in the highly loaded portions.

3. The lubricant for all main and connecting-rod bearings is supplied to the front of the crankshaft, which conducts and distributes the oil through machined passages. Fig. 11 shows the device used to introduce and de-aerate the oil.

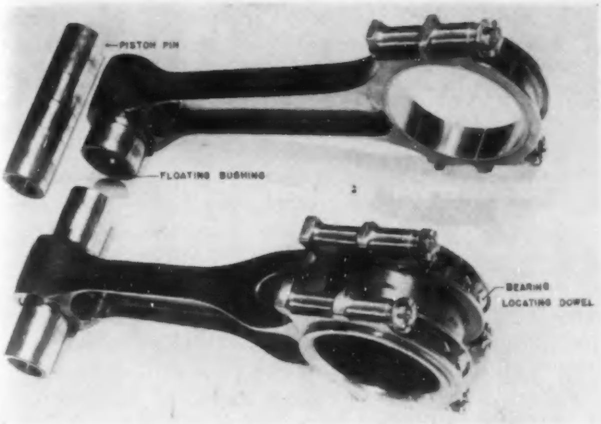
4. All crankpin and main journal oil outlets are fitted with orifice plugs, Fig. 12, which are expanded in place, and are employed to prevent the entrance of foreign matter into the bearings.

5. The lightening holes in the main journals and crank-

▪ Fig. 11 (right) – Crankshaft oil introducer



▪ Fig. 12 – Crankshaft oil metering orifice



■ Fig. 13 – Connecting rods

pins are barrel-shaped, to improve the strength/weight ratio of the shaft.

6. No crankshaft torsional vibration dampers of any type are used on this engine. The splines on the accessories driveshaft, which fit into the rear end of the crankshaft, were badly galled as a result of torsional vibration conditions. The calculated shaft frequencies show a single-node vibration at 137 cps and a two-node vibration at 357 cps. Wear on the crankpins, in the planes of the throws, was quite high. This condition could be explained in part by the copper/lead bearing materials employed.

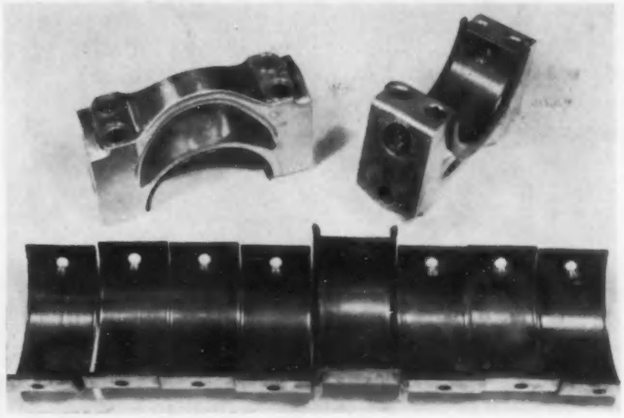
7. Little consideration of engine balance was evident. No provision for unbalance correction was found, and the blade and forked assemblies were installed at random as to bank. However, the measured rotary unbalance was quite respectable.

Connecting Rods – It will be observed in Fig. 13 that the connecting rods are of conventional form. Of interest is the floating cast-iron bushing at the pin ends, which is lubricated through the top of the eye. Although adequate, the weight added by this departure is considerable and its motive unknown. The clamped surfaces of the forked rod on the bearing were lubricated by oil from the bearing, which practice was of doubtful value because they were badly fretted. The bearing was doweled to the cap by a dowel inserted from the outside and wired in place.

Of interest also is the rod bolt locating slot in the outside, unstressed edge; the extreme closeness of the bolt holes to the bearing bore; and the excellent blending of all radii on the forked rod.

Bearings – The connecting-rod and main bearings were of 80-20 copper/lead, backed by very soft steel. The lead dispersion at the bond was good, but imperfect in the remainder. The main bearings, Fig. 14, are fastened to the caps with countersunk screws. No fretting was evident except on the center main, which was not so fastened.

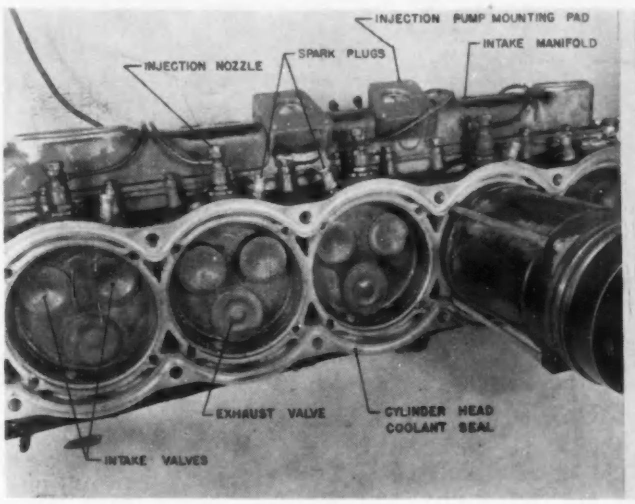
Cylinder Head – Fig. 15 shows the method of fastening the cylinder head and cylinder. Fourteen studs into the crankcase flange retain the head while four wet studs draw each cylinder barrel against a 45-deg seat on the cylinder head. A tin-plated copper shim is used on this sealing surface. A rubber packing at the bottom of the cylinder studs seals the coolant. A rubber gasket set in a groove



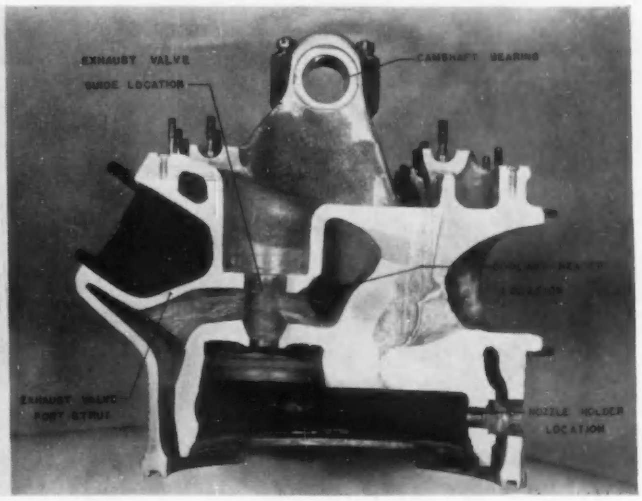
■ Fig. 14 – Main bearings and caps

in the head seals the crankcase-cylinder head joint. The top end of the barrel is sealed to the crankcase by two radial rubber seals with a vent to the outside between them.

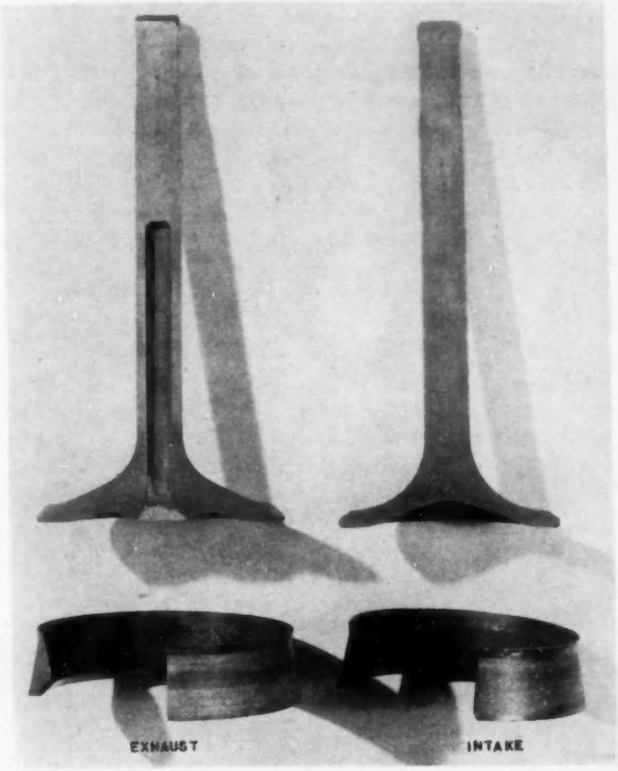
Fig. 16 illustrates a section of the cylinder head taken through the center of a combustion chamber, splitting the



■ Fig. 15 – Cylinder head and barrel assembly



■ Fig. 16 – Cylinder head cross-section



■ Fig. 17 – Valves and valve seats

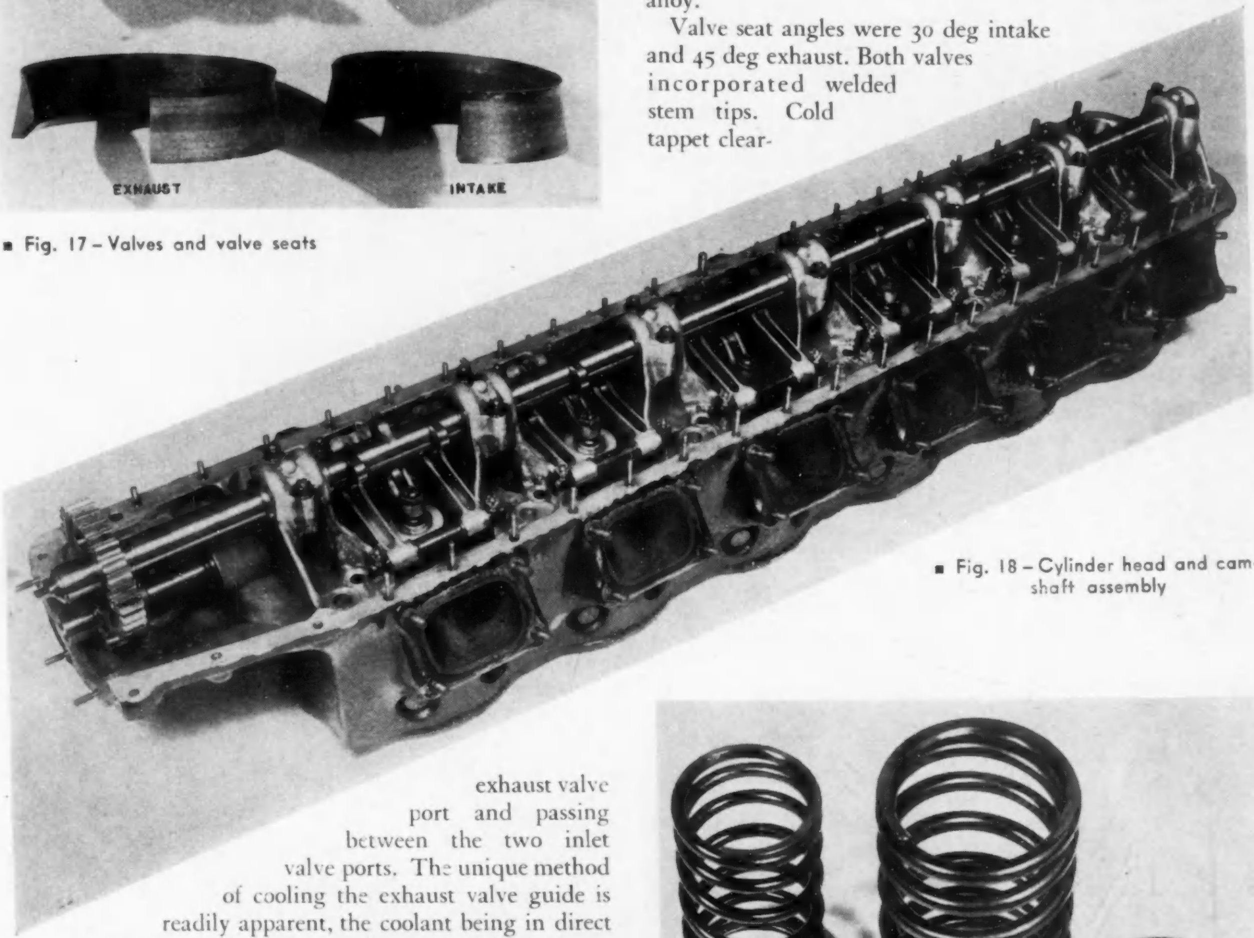
The cylinder barrel is surprisingly soft (245 Brinell hardness). It was honed, and the outside coated with a phenolic resin. Considerable lead attack was present on the inside surface. The fastening employed to hold the barrel and head together, consisting of the four wet studs fastened to lugs on the barrel, results in considerable bore distortion because of the bending locally applied. A bulge of the middle of the barrel of 0.010 to 0.015 in. when installed was noted, along with scuff tracks in the neighborhood of the lugs.

Coolant is supplied to the head by a long distributor tube which passes the length of the head through a cored passage (see Fig. 16). Holes in this header direct jets against all valve seats and the wet exhaust-valve guides.

Valve Gear – The valves are shown in Fig. 17. The exhaust valve is of an alloy corresponding to the usual austenitic valve steel and is faced with an alloy approximating Stellite No. 6 with iron dilution. Of three valves sectioned, the sodium had leaked out of one.

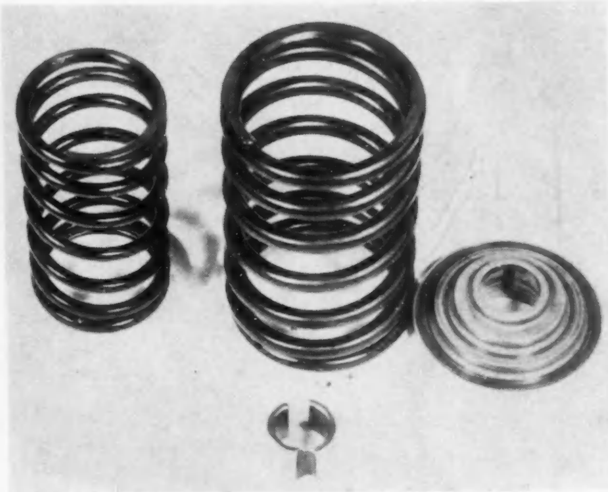
The intake valve is a 14% chromium martensitic alloy.

Valve seat angles were 30 deg intake and 45 deg exhaust. Both valves incorporated welded stem tips. Cold tappet clear-



■ Fig. 18 – Cylinder head and camshaft assembly

exhaust valve port and passing between the two inlet valve ports. The unique method of cooling the exhaust valve guide is readily apparent, the coolant being in direct contact with the exhaust-valve guide. To the lower right may be seen the location for the nozzle holder, also in direct contact with the cooling liquid. The struts, in both intake and exhaust ports, provide a very rigid design where coupled with the heavy wall sections of 1 and 1¼-in. thickness. The injection nozzles are all on the inside of the vee, and spray horizontally across the chamber. Four spark-plug holes are provided in each cylinder, of which only two are used, and the spark plugs are located for accessibility, being 180 deg apart in eight cylinders and roughly 120 deg apart in four.



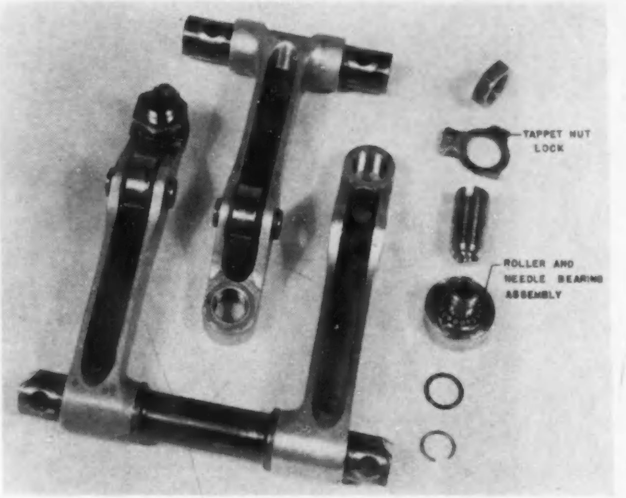
■ Fig. 19 – Valve springs, retainer, and keeper

ances ranged from 0.034 to 0.043 in. intake and 0.034 to .046 in. exhaust as received. Bronze valve guides are employed exclusively.

The valve port areas and valve open areas appear to be out of proportion. The intake port throat area is 6.34 sq in. while, with full lift of 0.68 in., the valve opening area is 8.35 sq in. Despite the high lift, all rubbing surfaces were in good condition. The velocity of air entering the combustion chamber is very low, due to the large port areas and the slow engine speeds, and probably accounts for the early intake valve closing of 36 deg ABC.

The camshaft, Fig. 18, is supported by seven large bearings of aluminum/iron alloy. Constructed of material equivalent to SAE X4340, the cams had a fine surface finish of 6-10 micro-in. rms and the shaft was bored for lightness. The estimated maximum grinding wheel diameter is 4.5 in.

A double spring assembly, Fig. 19, is used on each valve. These springs are of straight helical design rather than the



■ Fig. 20 – Rocker arm layout

tapered types usually associated with German engines. Loads are: valve closed – 56 lb, and valve open – 147 lb. The spring retainer locks are an unusual feature, consisting of a one-piece retainer of soft steel instead of the conventional two-piece design. This device may provide greater ease of assembly, and no fretting was found on the valve stems. Rocker arms are illustrated in Fig. 20, showing the roller needle bearing assembly of 17 rollers.

Pistons and Rings – The pistons, Figs. 21 and 22, are of forged aluminum alloy of more or less conventional design, employing 3 compression and 2 oil scraper rings. The piston-pin boss is lubricated by two diagonal holes from a groove below the upper oil ring. The insides of the pistons were sand-blasted, and no treatment at all was applied to the outside. The skirts are cam ground and quite free from scuffing.

Two-Speed Supercharger Drive – The two-speed supercharger drive, Fig. 23, incorporating a multiple-disc clutch with free-wheeling cam and slipper clutch, is adequately described elsewhere[3].

[3] See SAE Transactions, March, 1942, pp. 80-96: "Two-Speed Supercharger Drives," by F. M. Kincaid, Jr.

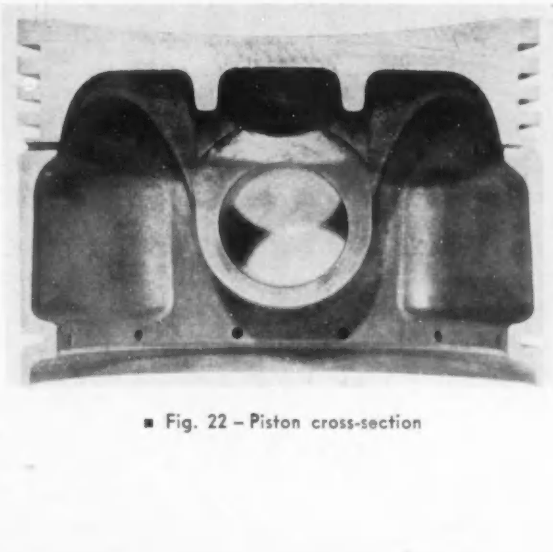
■ Fig. 21 – Pistons and rings

Supercharger – The supercharger housing and impeller are shown in Figs. 24 and 25. The fully shrouded impeller is a magnesium alloy forging, machined all over, a very difficult operation. Aluminum entrance vanes are used, but no diffuser. The performance of this unit is shown in Fig. 26.

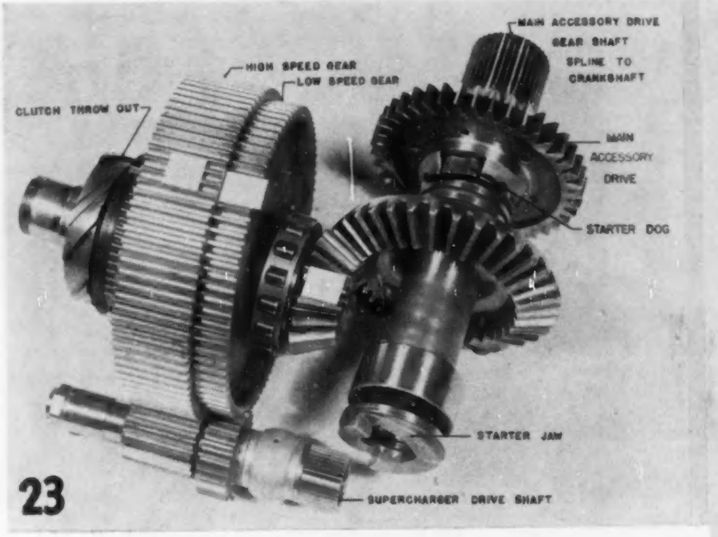
Ignition System – In common with domestic practice, two high-tension magnetos are provided, one supplying the exhaust-side plugs and the other the intake-side plugs. The outstanding feature of the system is the combination of automatic spark advance devices provided. When cranking, the magnetos are arranged to fire at 9 deg BTDC. As the engine speeds up, a centrifugal advance on the driveshaft advances the spark an additional 31 deg. This arrangement gives a spark advance of 40 deg for cruising. For boosted conditions, a throttle-operated switch retards the spark 15 deg, giving an advance of 25 deg for take-off and climb.

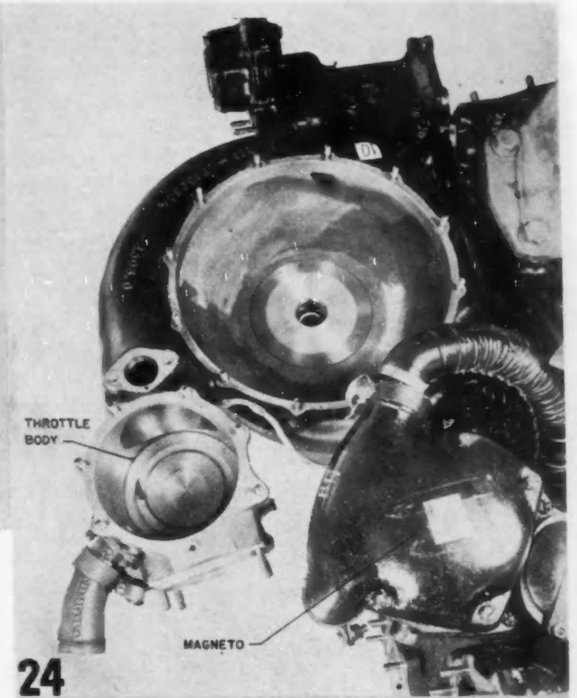
The means of obtaining this advance is of considerable interest. From the static timing, the first advance of 31 deg is obtained by a conventional centrifugal advance, employing small flyweights similar to current automotive practice. To obtain the retard from this point required for the take-off, each magneto contains an auxiliary breaker point. This second set, in parallel with the main points, goes to ground through a throttle switch.

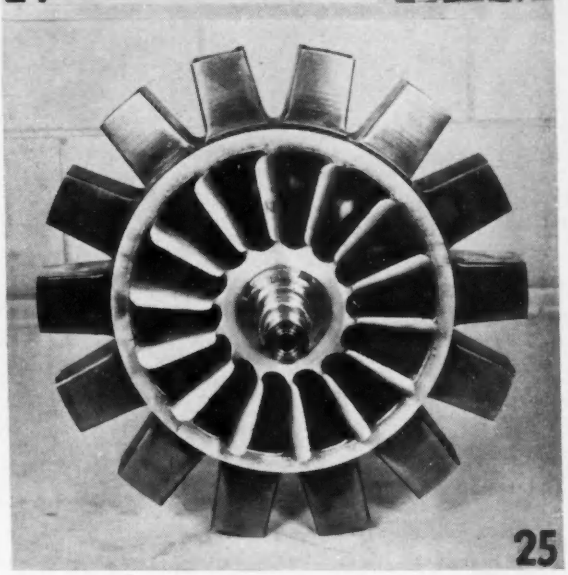
In normal operation the auxiliary points have no effect, as the throttle switch is open and no current flows. At take-off, the advanced position of the throttle linkage manually closes the throttle switch, causing the primary current to flow to ground through both sets of points.



■ Fig. 22 – Piston cross-section



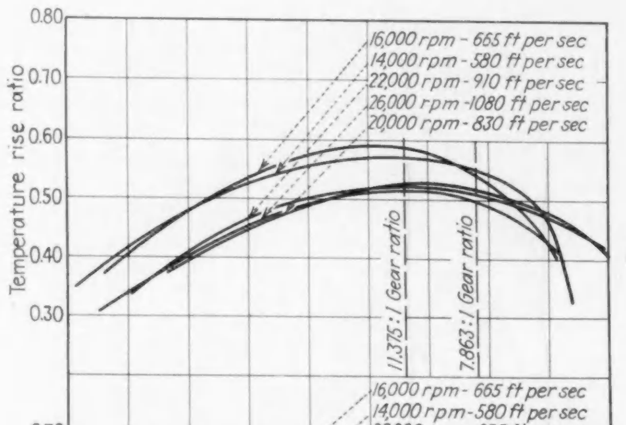




- Fig. 23 – Supercharger drive gears
- Fig. 24 – Supercharger housing and throttle body
- Fig. 25 – Supercharger impeller

Under these designated conditions, no spark occurs when the main points open, as the current is still flowing in the auxiliary system. The auxiliary points open 15 deg later, thus providing the necessary retard.

The magneto is a high-tension single-spark unit of the inductor type. That is, the cobalt magnets are stationary and four flux reverses per shaft revolution are produced by an inductor rotor. This design is probably older than the double magneto used on the Daimler-Benz.

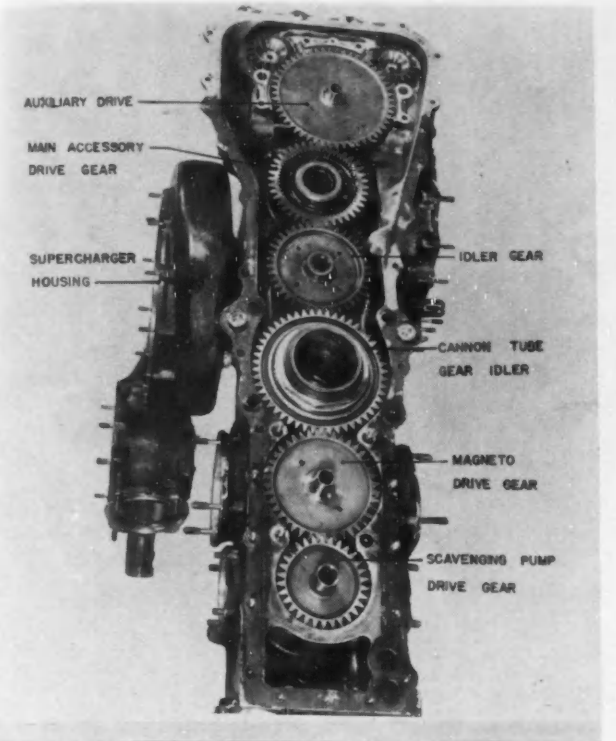


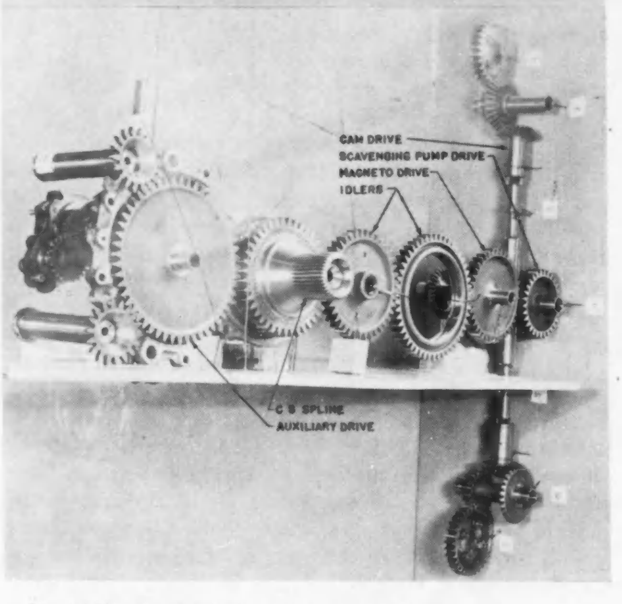
- Fig. 26 – Efficiency test of Junkers Jumo 211A engine supercharger

The shielding manifolds are of metal tubing with flexible connector shielding of simple metallic braid. No attempt is made to keep the system water-tight and clamps are used rather than threaded connectors. Connection to the spark plug is made by a tack stuck into the end of the wire, with rubber grommets between the sleeve and nut. The harness shows a surprising variety of materials, steel wire braid, brass and steel manifold tubing, steel hose clamps, tinned copper wire braid, aluminum marker tag, and elbow nuts, and brass connector nuts.

Beru mica spark plugs, having an imep rating of approximately 250 psi were used.

Gears and Splines – The spur gears have a relatively rough criss-cross surface pattern on the teeth, this surface evidently having been produced by the "Maag" process of





■ Fig. 27 (top) – Front view of accessory tower
■ Fig. 28 (below) – Accessory drive train

grinding. There is a reported acute shortage of grinding equipment for gears in Germany, and apparently the wheel feeds are set up rather fast to obtain a maximum number of parts per machine. The spur gears are of the 20-deg pressure angle, full-length type. The bevel gears are uniformly rough, apparently as a result of pickling. Whether or not this finish is intentional is conjectural.

The most noticeable feature of the splines used on this engine is that they are designed to do only the driving, centering means being provided elsewhere on the gears. A certain amount of backlash is therefore allowable, and consequently the splines may also be used as a small-angle universal joint, permitting a certain amount of misalignment. All splines are of the involute type.

A notable feature of the main reduction gear was the high polish in the radii at the roots of the teeth.

Accessory Drives – The accessory tower, Fig. 27, is made of cast aluminum and is attached to the rear of the crankcase. The foundry technique involved is very complicated. Assembly to the engine is convenient in that the only drive connection between it and the crankcase is the splined shaft which connects the crankshaft and main accessory drive gear. All of the gears have both their bearings in the accessory case so that their alignment is not disturbed by removing the rear section from the crankcase.

These features are offset, however, by the complexity of the gear trains (see Fig. 28). For example, it requires 30 gears and splines to drive from the crankshaft to the camshafts. Included in the accessories section are: *18 drives, which include:* 2 camshafts, 1 generator, 2 magnetos, 1 supercharger, 3 scavenging pumps, 1 oil pressure pump, 1 vacuum pump, 1 tachometer, 1 starter, 1 injection pump, 1 fuel transfer pump, 1 coolant pump, and 2 unused auxiliary drives.

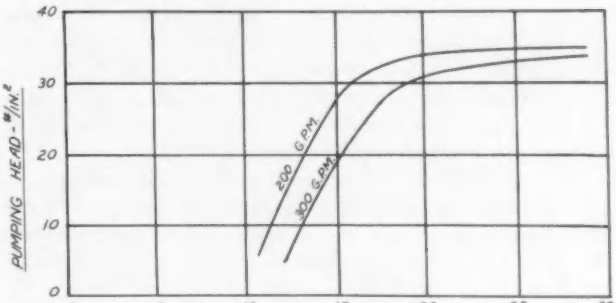
Curiously, no timing marks could be discovered on any engine part.

Starter and Generator – The starter, Bosch Series 11, is an inertia flywheel type. Flywheel acceleration is accomplished by a small four-pole series motor designed for 24 v. Rated speed is 12,000 rpm, at which speed the energy storage in the flywheel is calculated at 3490 ft-lb. Fourteen sec are required to accelerate to this speed. An unusual feature is the brush-lifting mechanism. This is a manual control and lifts all four brushes from the commutator to remove brush friction during hand cranking.

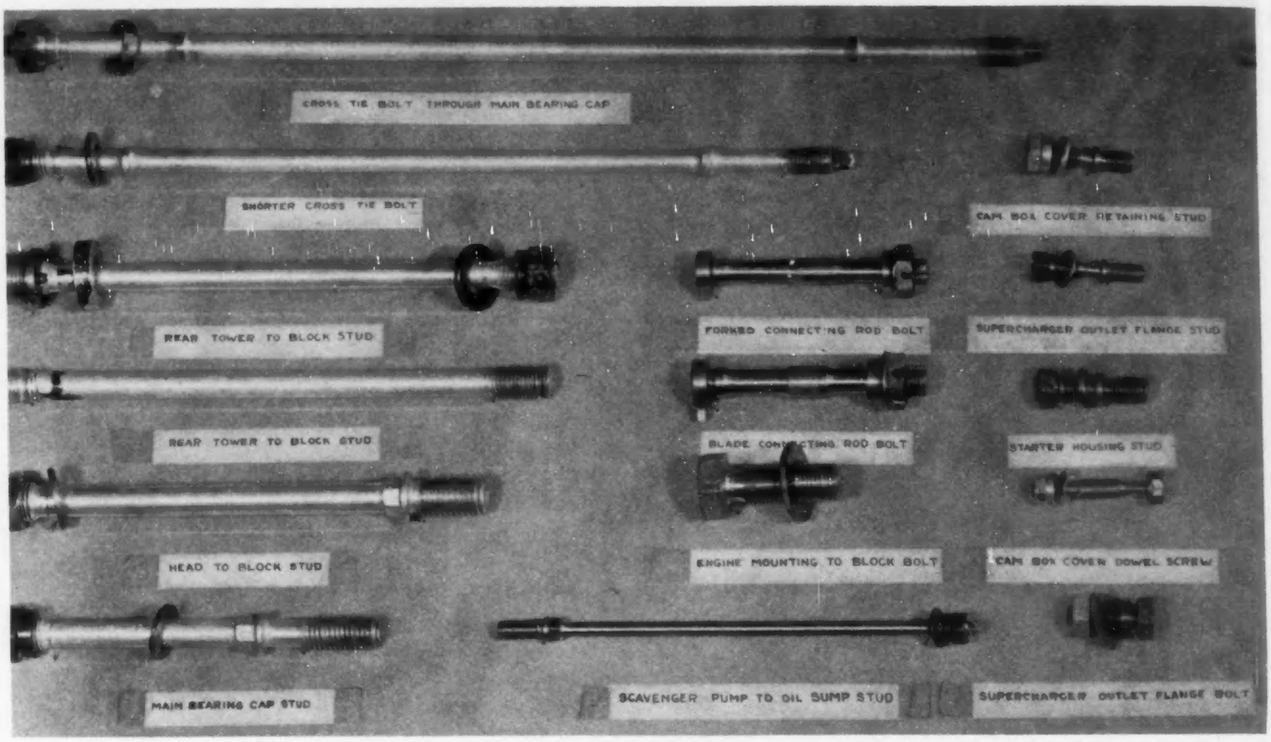
The 24-v generator is a four-pole shunt-wound machine with forced cooling over the commutator only. A quill shaft drive of ⅜-in. diameter by 8 in. is used for vibration isolation.

Coolant Pump – The coolant pump is of the centrifugal type with a double volute outlet and a fully shrouded impeller. Its performance showed it to be designed for size rather than efficiency, which was only 53% at the estimated operating point. The sharp elbow at its inlet produced considerable cavitation and its altitude performance was mediocre, as is shown in Fig. 29.

Oil System – The oil pressure pump consists of steel gears running in a magnesium housing. There are eight teeth on each gear. Full pump capacity was 12 gpm at 45 psi pressure, the pressure at which the relief valve was set when received. The pump is attached directly to the



■ Fig. 29 – Coolant pump cavitation curve – 3070 rpm, ethylene glycol at 230 F



■ **Fig. 30 (above and continued on opposite page) – Threaded fastenings**

strainer housing, with a check valve on the pump outlet to prevent loss of prime. Outlet oil passes directly through the disc strainer, which is kept clean by rotation of the discs past stationary blades. This motion is supplied by a ratchet from the throttle linkage, so that any change of the throttle cleans the filter.

The engine contains six scavenging pumps built in three assemblies of two each. Each cam box has a double scavenging pump located at the rear end, and a pair of main scavenging pumps are located in the accessory tower. The cam box scavengers discharge their oil to the crankcase lower cover, where it is picked up by the main scavenging pump along with the power section oil and delivered to the oil coolers and tank. These pumps are all straight spur-gear type.

Oil flow is directed from the oil pump and strainer assembly through cored passages to the rear gear tower and thence outward through an external line to the main reduction gear case. A tee at the approximate center of the line supplies oil to the fuel injection pump. A second tee mounted near the reduction gear case diverts oil through two lines to the cam boxes. Internal piping within the reduction gear case conducts oil to a jet for the main reduction gear, and through a pressure line and cored passage to the centerline of the crankshaft.

Seals and Gaskets – The Jumo 211B is rather unique in two respects regarding sealing practice: First, there are few metal-to-metal sealing surfaces, paper and synthetic rubber gaskets and seals being used extensively. Second, synthetic rubber gaskets mounted in grooves are used in great numbers, both as coolant and as oil seals.

The predominant type of synthetic rubber used is Buna S, which is impervious to oil and ethylene glycol and ages fairly well. Two parts were of Buna N and two of natural rubber. Hardness varied from 55 to 70 durometer.

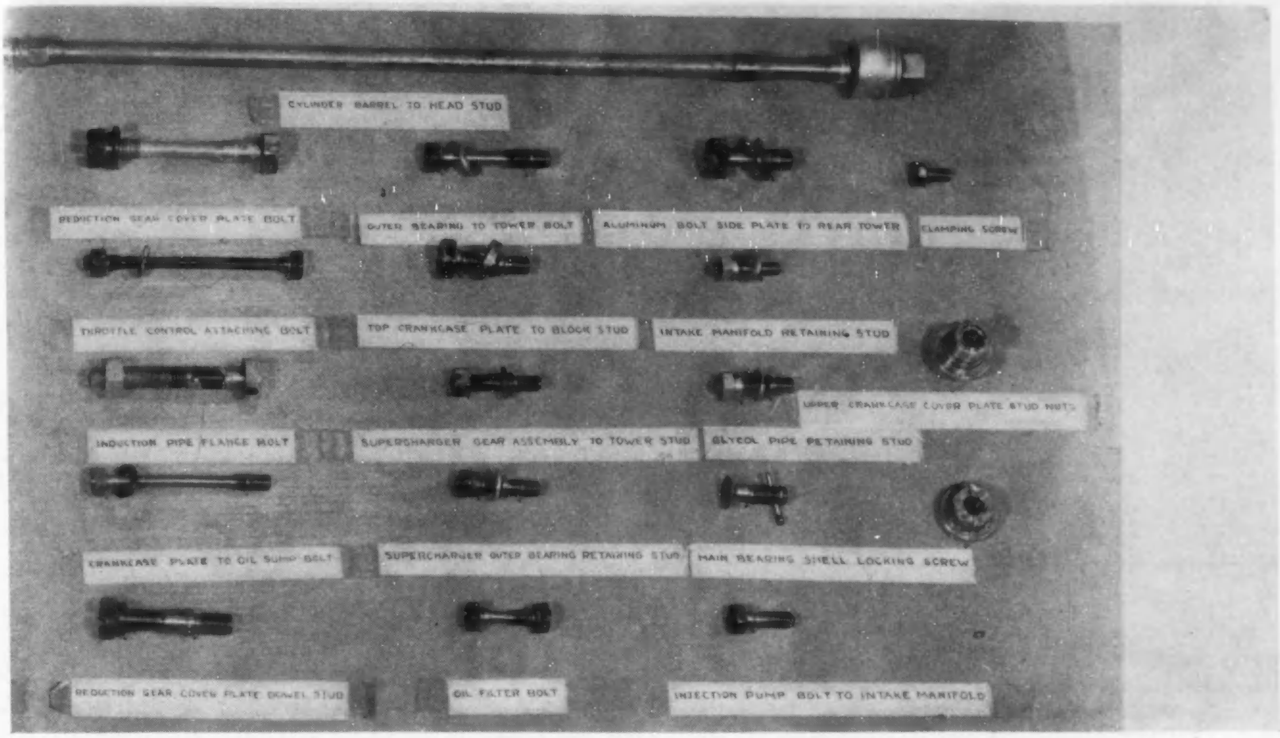
Threaded Fastenings – Threaded fastenings of the Jumo 211B, Fig. 30, have proved to be a source of interest. No evidence of ground threads has hitherto been reported in German designs. On this engine, however, microcomparator photographs of a representative sample of small highly loaded stud bolts indicate that the threads are ground. A good degree of uniformity of pitch, pitch diameter, flank angle, and particularly root radius, is maintained.

Reduction of bolt weight is effected by taper counterboring the hexagonal heads of cap bolts, and the use of aluminum bolts in a few light load applications. Stress concentrations are minimized by applying generous fillets which undercut the root diameter of the threads and by the use of long tapers as a means of blending the thread section to the small diameter shank. Cadmium plating is used extensively. As received, the bearing cap stud stress was 117,000 psi and that of the rod bolts checked at 90,000 to 96,000 psi. A large number of flanged studs were used, which were secured very tightly against the flange, but employed no interference fit on the thread. Soft nuts were used extensively, including many of aluminum, to distribute the thread load.

Materials – In general, the analysis of materials uncovered no new metallurgical art.

The lavish use of alloying elements in the steels and the wide variation of chemistry in parts requiring similar properties are apparent from a study of Appendix III. The cleanliness of the steel was exceptional, but the control of grain size appears to have been quite neglected. Carburizing was good as to depth, but an excess of massive carbides was evident.

A substantial amount of magnesium was found, including the supercharger housing, reduction-gear cover, and

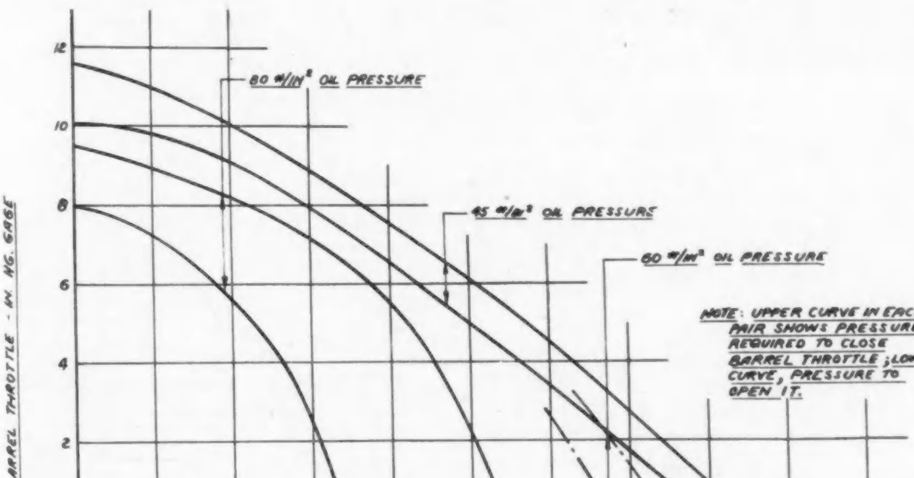


piston-pin plugs. This condition contrasts with the Daimler-Benz DB 601-A in which little magnesium was used.

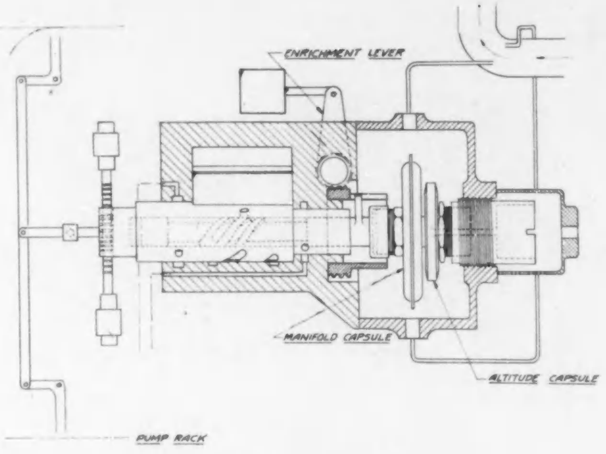
Most of the aluminum alloys have high silicon content. The 0.63% cobalt in the cylinder head is notable particularly since not present in the other, presumably lower stressed, aluminum castings.

Boost Control and Throttle Unit – This device is installed between supercharger and engine, and is essentially a barrel-type throttle, manually controlled through differential gearing such that the hydraulic unit may override the manual control. The automatic hydraulic unit consists of:

1. A sylphon control unit, subjected to manifold pressure and operating an oil bleed valve.



■ Fig. 31 – Boost control regulation curves



■ Fig. 32 – Mixture control diagram

2. A servo piston subjected to an engine oil pressure of 45 psi through a restriction. The piston is connected to the barrel throttle through the pinions of the differential.

In operation, excess manifold pressure on the sylphon closes the bleed valve. The oil pressure, thus built up, moves the servo piston against the action of the piston return springs, and thereby, through the differential gearing, closes the barrel throttle sufficiently to establish equilibrium.

The set pressure of the device is controlled by the position of the manual throttle through a mechanism which moves the base of the sylphon unit loading spring. At wide-open manual throttle, the set pressure was 10¾ in. hg, the unit controlling between 10 and 11½ in. hg (see Fig. 31), with a regulation droop of about 1½ in.

The minimum operating oil pressure is 30 psi, since this pressure is necessary to overcome the force exerted by the piston return springs.

An adjustment button is provided to change the maximum boost setting.

The device responds rapidly and has little or no tendency to hunt, a characteristic of proportioning regulators with not too great sensitivity.

In case of mechanical failure of the manual control means, a spring pulls the manual control to a wide-open-throttle position.

In case of oil pressure failure, the servo piston springs move the barrel to a wide-open-throttle position.

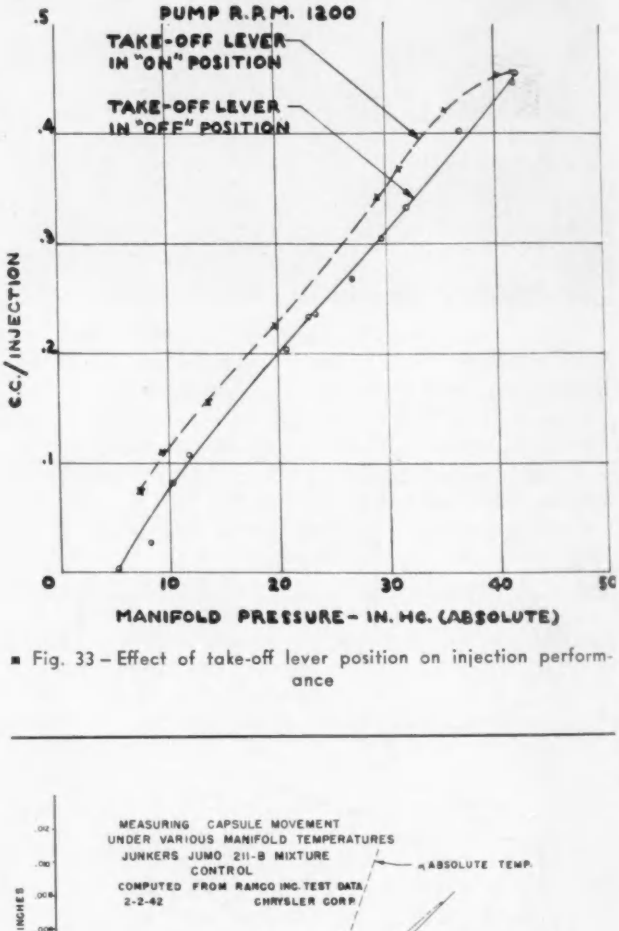
The complete control weighs 8.55 lb, and is made up of 215 parts. The larger parts are made of a magnesium alloy. Fiber bearings are used at many points. Oil from the servo unit drains to the cam box.

The design is ingenious and the unit well made, although the construction seems expensive for an essentially protective device.

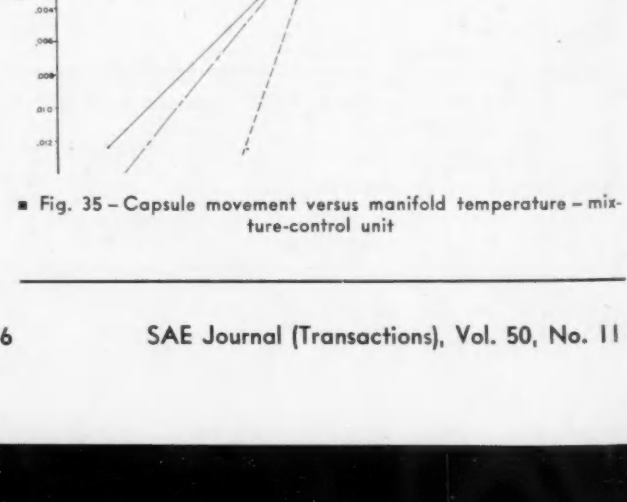
Mixture Control Unit – This unit, Fig. 32, mounted directly in the injection pump, is classed as a pressure-temperature type control.

Pressure- and temperature-sensitive capsules move the servo valve, valving oil to a rotary vane type servo which, through suitable linkage, controls the injection pump rack. Two capsules form the basic control, one being sensitive to manifold pressure and temperature, the other to the difference between manifold and atmospheric pressure. Both move the servo pilot valve axially. The pilot valve has helically cut valving lands so designed that both rotary and axial movement produce rotation of the servo motor and its connected parts. An enriching lever is provided which rotates the servo valve. This lever is probably controlled manually to lean the cruise mixture, and also by a solenoid energized through a switch, probably operated from the manual throttle. A logical arrangement would be to advance the pump rack for both take-off and idle mixture enrichment.

The advantages of this system of mixture control seem



■ Fig. 33 – Effect of take-off lever position on injection performance

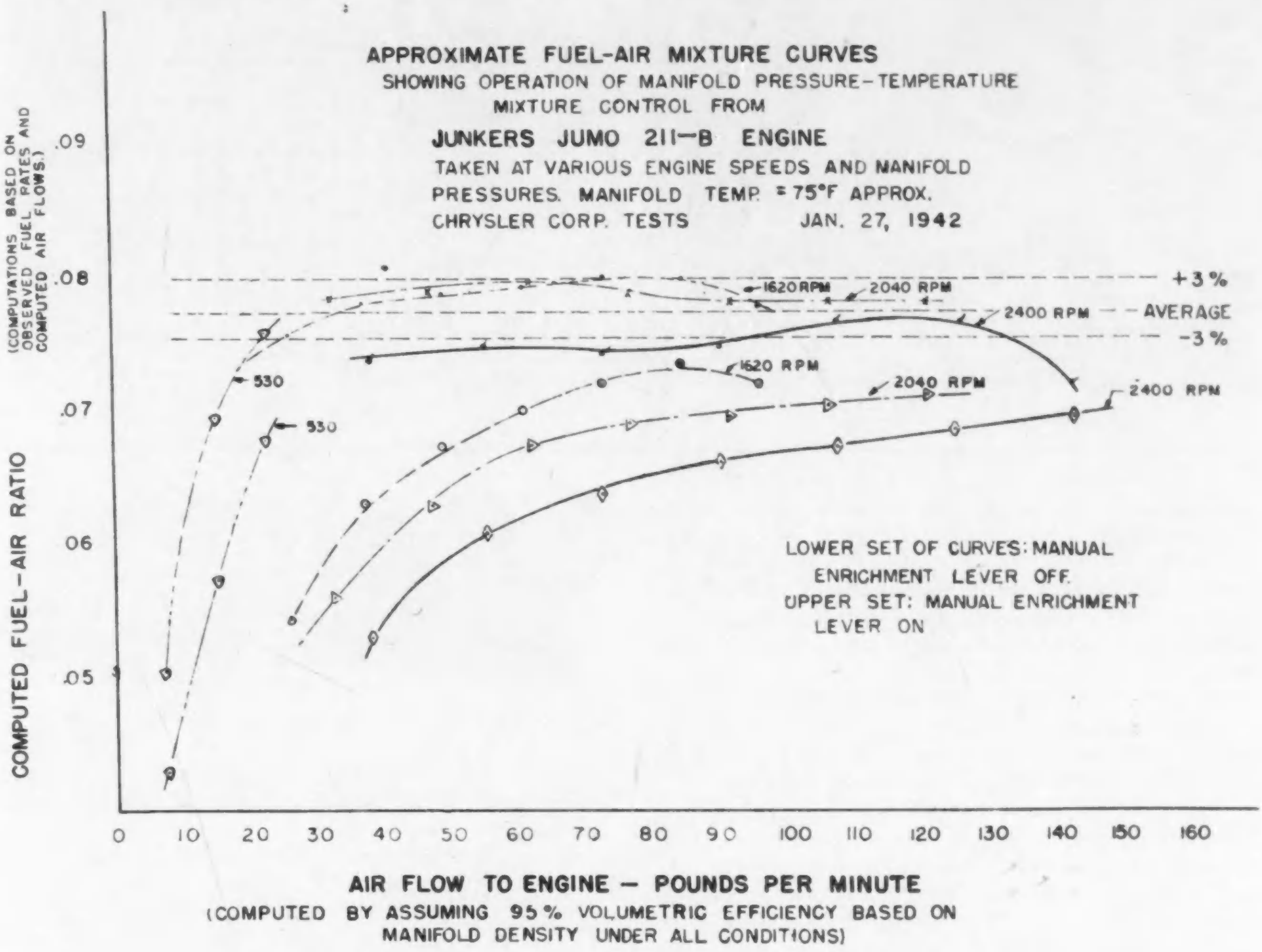


■ Fig. 35 – Capsule movement versus manifold temperature – mixture-control unit

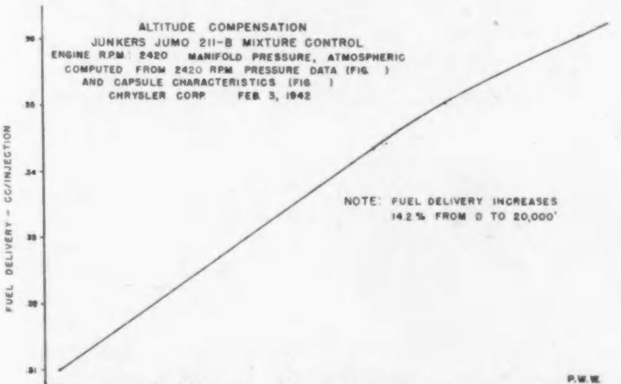
to be its relatively light weight of 9.6 lb, its compactness and simplicity, the unit being made up of 125 different parts, exclusive of screws, lock washers, and so on.

Its disadvantages are: poor control of the mixture at cruise or light-load conditions with the setting as received, and lack of safety precautions in case of failure.

Air flow through the capsule housing is provided by means of two tubes – one being subjected to total manifold



■ Fig. 34 – Calculated fuel-air ratio curves showing operation of mixture-control unit



■ Fig. 36 – Effect of altitude capsule on fuel delivery

pressure, the other, in addition to being located at the inside of a bend, is subjected to static, minus velocity pressure. Air velocity through the capsule housing is not a significant factor provided it is sufficient to keep the temperature capsule at manifold temperature.

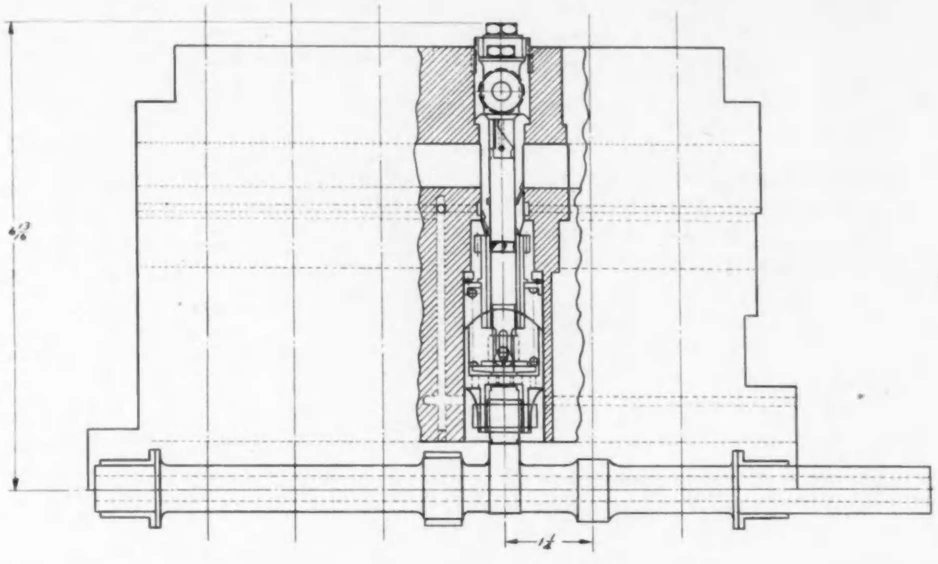
No acceleration means are incorporated. Several adjustments are provided.

The entire capsule unit may be shifted axially by means of an adjusting screw accurately marked. This adjustment moves the entire delivery versus manifold pressure curve up and down.

It is possible to change the slope of this curve to any reasonable value by moving a pivot location which changes the ratio between servo motor and rack movement.

In the event of oil pressure failure of one of the capsules, there appears to be no manual means of controlling the mixture ratio. Failure of oil pressure leaves the mixture indeterminate. Failure of a capsule causes the mixture to become lean. A built-in gear pump makes oil pressure failure less likely than with a remote pump.

Fig. 33 shows the effect of the enrichment lever position



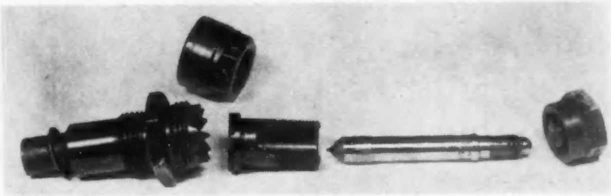
■ Fig. 37 – Fuel injection pump

on the pump delivery-manifold pressure relation at 1200 pump rpm. Since the enrichment lever in effect rotates the servo valve 12 deg, the result is an increase in average pump delivery of 0.037 cc per injection under all speed and manifold conditions. If the enriched curve is of constant slope and intersects the origin, a substantially constant mixture ratio will result. However, with the enrichment lever in the lean position, the curve is shifted downward 0.037 cc per injection. The slope is still that of the enriched curve but no longer intersecting zero; it therefore, is impossible to secure constant mixture under this condition.

Fig. 34 of calculated fuel-air ratio under several conditions of operations indicates very clearly this tendency to lean off at low manifold pressures.

The fundamental disadvantage of the pressure-temperature type control is indicated here, in that no direct indication of engine speed or air or fuel velocity is available. Thus, any required change in mixture ratio, which is a function of engine horse power or air velocity rather than torque or manifold pressure, is not readily obtained. When the engine speed-air weight relation is not linear or the manifold pressure-air weight relation is not a continuous curve, compensation difficulties may be expected.

Test data indicate that the temperature capsule is so



■ Fig. 38 – Injection nozzle

designed that fuel delivery becomes approximate inversely proportional to the square root of the manifold temperature.

Fig. 35, showing temperature capsule deflection versus temperature, as tested while removed from the control, indicates an exponential relation of slightly less than 0.5. Exponential relations of 1 and 0.5 are dotted for reference.

Fig. 36, a plot of the effect of the altitude capsule, indicates an increased fuel delivery of 14.2% from 0 to 20,000-ft altitude or an average of about 0.71% per 1000 ft. This value is considerably higher than the 0.29% per 1000 ft secured on a DB tested in this country.

Mixture Control Appraisal –

A. The control is exceptionally light, compact, refined in

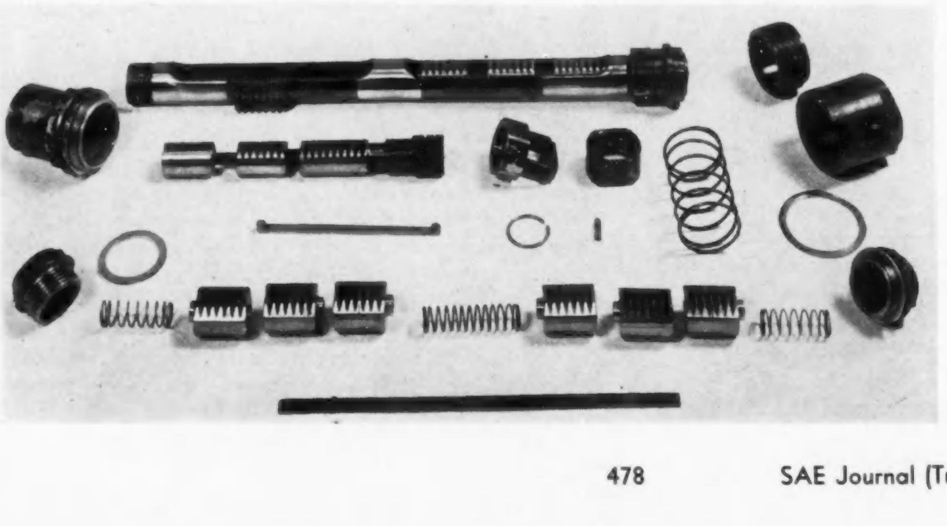
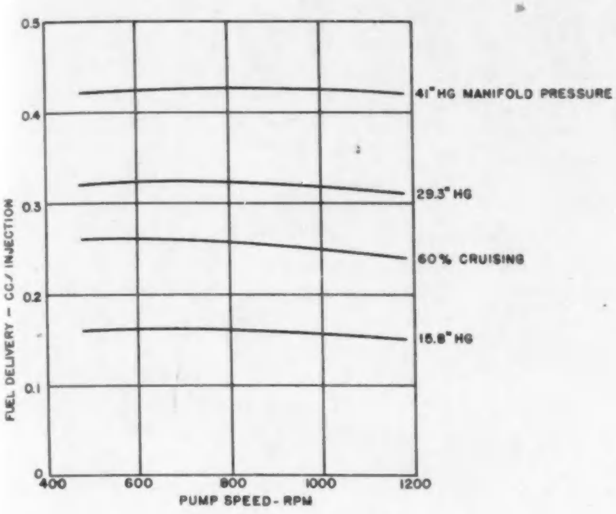
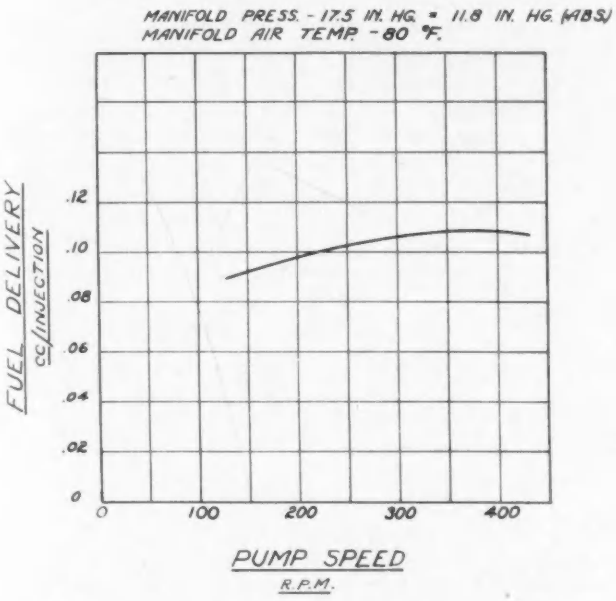


Fig. 39 – Injection pump control rack assembly



■ Fig. 40 – Injection performance



■ Fig. 41 – Idle-range performance – injection equipment

design, and indicates workmanship of very high grade.

B. The performance of the control is quite acceptable in the high power and idling range with the assistance of the enrichment lever.

C. The performance of the control under cruising conditions does not seem to be satisfactory with the setting received.

D. The control is designed very close to the limit of the engine performance. A major increase in rating would necessitate a redesign.

E. The control has no provision for manual operation in case of oil failure.

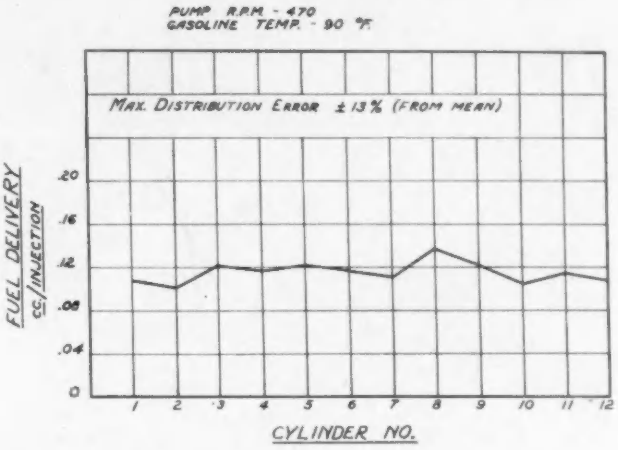
F. As received the control was not set to provide a mixture corresponding to full-rich for take-off.

G. Test results could be closely repeated, indicating precise action of the control.

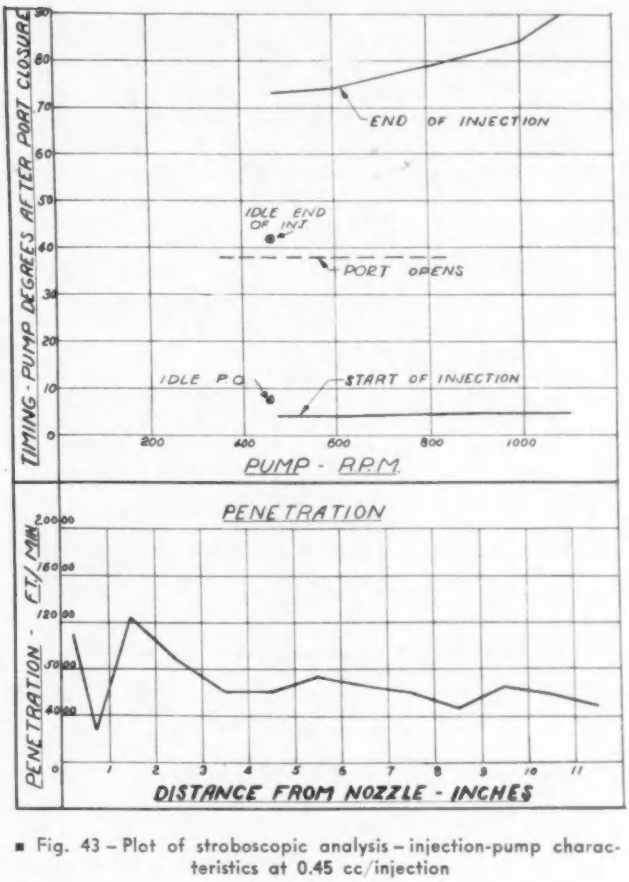
Injection Pump – The injection pump, Fig. 37, is of the standard Bosch diesel-type construction, having a 9-mm bore, 8½-mm stroke, 150-deg inverted V-12 arrangement, and mounted in the engine V by means of flexible metallic mountings.

The following features are of interest:

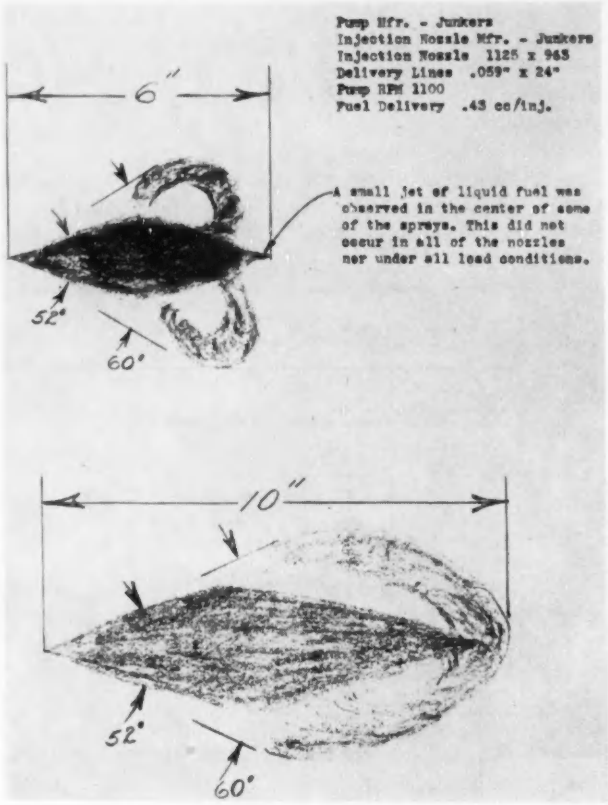
1. Well-designed plunger and barrel construction, including offset-shaped ports for fast porting; also plunger lubrication and sealing.



■ Fig. 42 – Distribution under no load and 60% cruising conditions – injection equipment performance



■ Fig. 43 – Plot of stroboscopic analysis – injection-pump characteristics at 0.45 cc/injection



■ Fig. 44 – Nozzle spray patterns

2. Convenient means of external, individual plunger adjustment, for calibration.

3. Dual series delivery valves.

4. Cam contour providing fast pumping stroke with slow return.

Injection Equipment Details – The total weight of the system, including injection pump, vapor trap, mixture control, lines, nozzles, and transfer pump is 61.84 lb.

The injection nozzles, Fig. 38, are of the open, single-orifice, centrifugal type, without moving parts. The fuel channel contacts the nozzle body at two points only, thus providing heat insulation.

All delivery lines are 0.059-in. bore and of lengths between 16 and 24 in.

The transfer pump is of a two-piston, oscillating-cylinder design, valving by means of the oscillating cylinders, and regulated by means of a bypass.

The vapor trap consists of a relatively large curved chamber, well baffled and equipped with float to valve off vapor. This unit is located between the transfer and injection pumps.

The only fuel filtering means included was the fairly coarse screen located in the vapor trap, it being reasonable to assume that a more effective strainer was applied at some other point.

The injection pump control rack assembly is shown in Fig. 39.

The uniform slopes shown in Fig. 40, run at the various rack positions, indicated fundamental stability of the injection system. The downward slope with increased speed indicates minor plunger leakage and adequate porting.

Fig. 41 of idle range performance shows a drop-off with decreased speeds, also indicative of minor plunger leakage.

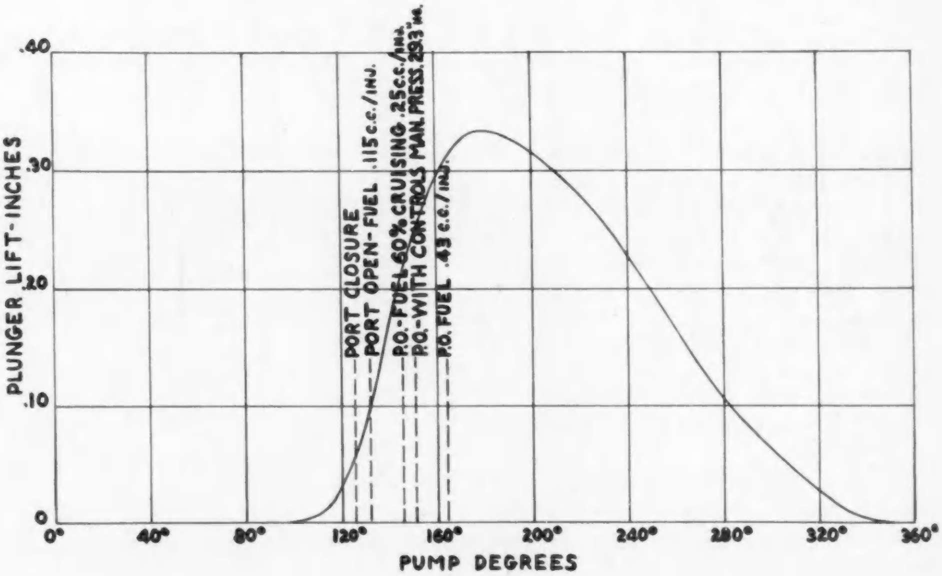
Data indicate that the pump was relatively insensitive to transfer fuel temperature and pressure.

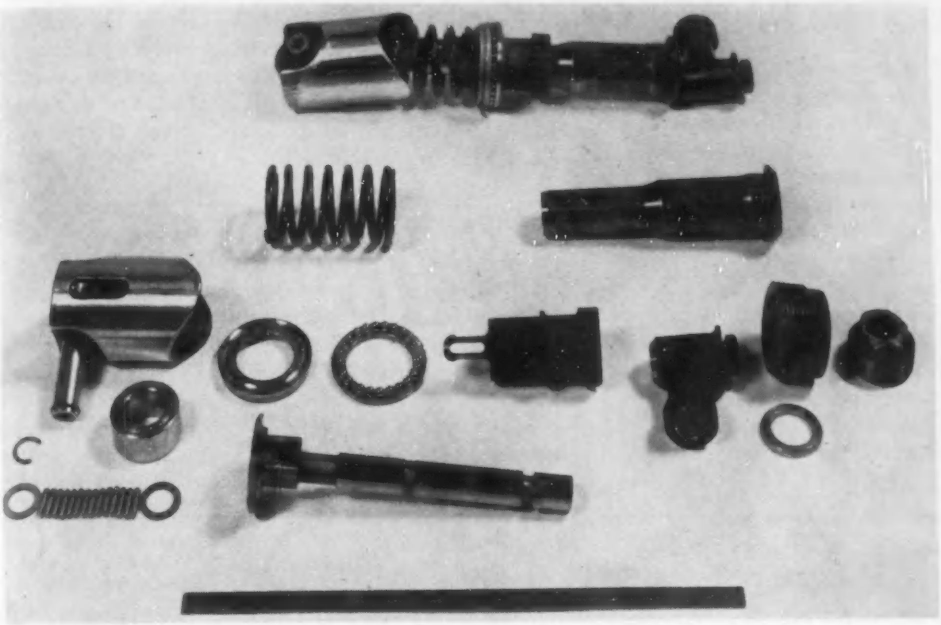
From Fig. 42, the maximum distribution error is noted to be 13%, and without the worst cylinder, 9%. This test was conducted at the relatively low pump speed of 470 rpm and a fuel delivery of 0.1 cc per injection or about 25% of full fuel delivery.

Fig. 43, a plot of stroboscopic analysis, indicates the following:

The start of injection is constant with respect to speed and load, thus giving a constant lag of low value. The end of injection increases normally with speed and reaches a maximum value of 184 deg crank angle. With pump port closure timed 76 deg ATC, end of injection occurs 80 deg

■ Fig. 45 – Injection equipment performance





■ Fig. 46 – Injection pump cam follower and plunger

ABC. This function is late for best fuel consumption, but occurs only during maximum power conditions. At 60% cruising, the end of injection occurs 12 deg ABC.

At idle fuel rates the spray velocity is higher, due to a lower degree of atomization. The pattern is shorter, due to the smaller quantity of injected fuel.

Fig. 44 indicates the cone angle to be constant over the speed and load range.

Visual stroboscopic examination showed regularity of successive injections. Atomization is good, the start of injection being sharp and well defined.

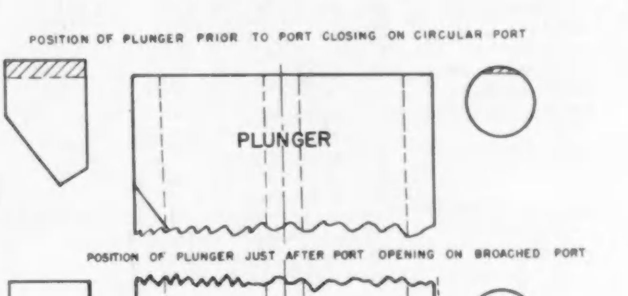
The Junkers nozzle installation would probably give a maximum line pressure of 2000 to 3000 psi.

Fig. 45 shows the plunger lift-cam position relation, as well as port functions at several fuel rates. The slow plunger return rate may make possible the relatively light tappet springs found in this pump. Fig. 46 shows the injection pump cam follower and plunger.

Fig. 47 indicates the porting arrangement. The plunger, in addition to the double helix, is cross and axially drilled, possibly to permit rapid pressure equalization. In general, the porting arrangement may produce a general flow across the fuel gallery, that is, from right to left in the diagram.

Pressure lubrication is provided to the lower barrel annulus. The upper barrel annulus pressure is limited to that of the transfer pump due to its communication with the fuel gallery. Since lubricating oil pressure is probably higher than transfer pump pressure, the general direction of oil flow will be from the lower to the upper annulus. The plunger under-cut may assist in oiling the lower portion of the plunger barrel assembly. Greatest plunger wear was noted on that area swept by the lower barrel annulus.

Oil is supplied by the built-in pump, through a check valve to longitudinal galleries which communicate with the lower barrel annulus. An additional passage runs to the mixture control servo valve. A relief valve in this passage permits bypassed oil as well as oil discharged from the low pressure side of the servo vane to return to the cam case, and out through the hollow driveshaft. No filters were apparent in this circuit.



■ Fig. 47 – Schematic diagram of plunger and barrel porting and helix arrangement – fuel injection pump

■ Potential Development of the Jumo 211

From available data, it is estimated that the Junkers 211B engine delivered approximately 1100 bhp, giving it a specific weight at take-off of 1.37 lb per bhp, a mediocre value. To increase the output through higher mean effective pressures would involve changes of such inconvenience that it appears doubtful whether this possibility was seriously considered in the design. The injection pump is operated at full capacity, while augmenting the fuel rate would require spreading plunger centers. The supercharger is already operated beyond efficient tip speeds for its type. No provision for crankshaft torsional vibration dampers was made, while a badly fretted accessories drive spline already shows symptoms of the disease. The valve and spark plugs approach their mep limit. Wall thicknesses of the cylinder head are light in section limited by the cylinder center distances. Finally, 92-octane gasoline, the apparent German available grade, is probably approached by the present rating, estimated at 210 to 215 psi imep.

Greater rotative speed is the evident channel of development for this engine, but its 6½-in. stroke may impose a few problems in the way.

■ Acknowledgments

The generosity of Henry Timken in securing this engine for analysis is greatly appreciated. Calibration of the supercharger was made by the General Electric Co.; of the altitude capsule by the Ranco Co.; of the spark plugs by the Ethyl Gasoline Corp.; and analysis of some of the materials was made by the Timken Roller Bearing Co. The efforts of a large group of engineers of the Chrysler Engineering Laboratories are responsible for the remainder of the work.

Appendix I

Engine Data

Name	Junkers Jumo
Model	211B
Type	Inverted 60-deg V, liquid-cooled Otto-cycle, fuel injection
No. of Cylinders	12, in two banks of 6 each
Bore	5.904 in.
Stroke	6.490 in.
Piston Displacement	2132.1 cu in.
Compression Ratio	6.56:1
Reduction Gear	Spur-gear type 32:10 (crankshaft / propeller shaft) ratio
Direction of Rotation	(From anti-propeller end) Crankshaft – counterclockwise Propeller shaft – clockwise
Valve Gear	Underhead camshaft with rocker arms actuating two inlet and one exhaust valve per cylinder
Firing Order	1, 9, 4, 11, 2, 7, 6, 10, 3, 8, 5, 12
Valve Clearance (cold)	24 Intakes 0.033 to 0.042 in. (mean: 0.037 in.) 12 Exhausts 0.034 to 0.046 in. (mean: 0.040 in.)

Engine Data (Continued)

Valve Timing	Intake Opens	12 deg BTC
	Intake Closes	36 deg ABC
	Exhaust Opens	64 deg BBC
	Exhaust Closes	18 deg ATC
Valve Lift (zero lash)	0.7084 in.	
Supercharger gear ratio	Low speed	7.86 x crankshaft
	High speed	11.38 x crankshaft

Engine Dimensions

Length	– tip of starter to end of propeller shaft	86¾ in.
	end of cam box to end of propeller shaft	69¾ in.
	cam box length	48 in.
Width	– tip to tip of exhaust stacks	53½ in.
	to edge of cam boxes	31 in.
Height	–	42½ in.

Weight – 1678 lb complete, as received
1510 lb rating weight

Manifold Pressure – 41½ in. hg at W.O.T. (from test)

Power Output – Not checked on this engine. Most sources give 2400 rpm as take-off speed. The manifold pressure control settings of 41½ in. hg abs., indicate about 170 bmep or 1100 bhp. Flight, Jan. 18 1940, gives 1050 bhp at 2300 rpm.

Comparison of Engine Specifications and Performance*

1. Make	Mercedes Benz	Allison	Rolls Royce	Hispano Suiza	Jumo
2. Model	DB-601A	V-1710C-15	Merlin X	12Y-51	211B
3. Number of Cylinders	12	12	12	12	12
4. Arrangement	Inverted V	V	V	V	Inverted V
5. Bore, in.	5.7	5.5	5.4	5.9	5.9
6. Stroke, in.	6.3	6.0	6.0	6.7	6.5
7. Piston Displacement, cu in.	2070	1710	1647	2197	2132
8. Military Rating, hp	1000	1090	1025	1100	975
9. Military Rating, rpm	2400	3000	3000	2400	2300
10. Military Rating Altitude, ft	14760	13200	17750 (High Blower)	10696	
11. Hypothetical Horsepower at 15000 ft	990	1020	1150	920	990
12. Take-Off Rating, hp	1150	1040	1045	1100	1100
13. Take-Off Rating, rpm	2500	3000	2850	2400	2400
14. Bmep (Military Rating), psi	158	168	164	156	157
15. Bmep (Take-Off), psi	167	160	176	166	170
16. Compression Ratio	6.8	6.65	—	7.0	6.56
17. Take-Off Piston Speed, fpm	2625	3000	2850	2690	2600
18. Total Piston Head Area, sq in.	306	285.5	275	328	328
19. Take-Off Horsepower per Square Inch Piston Area	3.84	3.65	3.81	3.36	3.36
20. Take-Off Horsepower per Cubic Inch Displacement per Minute	0.000111	0.000101	0.000111	0.000104	0.000107
21. Dry Weight, lb	1367	1325	1394	1085	1510
22. Dry Weight per Unit Displacement, lb per cu in.	0.66	0.775	0.845	0.493	0.708
23. Unit Weight, lb per take-off hp	1.19	1.27	1.33	0.995	1.37
24. Height, in.	40.5	42.1	41.1	37.2	42.5
25. Width, in.	29.1	30.6	29.8	30.1	31.0
26. Overall Length, in.	84.0	94.5	75.1	84.1	69.7

*All data excepting that for the Junkers Jumo 211B are reproduced from Reference (1).

Appendix II

Engine Weights

Part	Lb	Oz
Crankcase (bare)	232	0
Crankcase upper coverplate	18	4
Crankcase lower coverplate	6	9¼
Two cam-box and head assemblies including cam-box covers	321	0
Eight main crankshaft bearings and caps	21	5¾
Twelve piston and rod assemblies with bearings	149	4½
Rear assembly with gear train	109	4½
Supercharger assembly	40	0
Supercharger controls and throttle	11	10¼
Oil pump and filter	13	7¼
Reduction gear and shaft assembly (without propeller shaft)	71	7¼
Two magnetos and line assemblies	42	8
Crankshaft	164	1
Crankshaft gear	16	12
Crankshaft gear nut, tapered rings, and locks	2	6
Primary and secondary fuel pumps	60	0
Twelve cylinder barrels	105	6
Water pump assembly	10	14¼
Miscellaneous lines, tubes, nuts and brackets	113	7
Rating Weight	1509	11

Parts not included in Rating Weight:

Part	Lb	Oz
Four engine mountings	24	0
Generator	25	0
Starter	34	12
Exhaust stacks and cowling brackets	67	8
Propeller shaft	16	13
Total Weight	1677	12

	Lb	Oz
1. Cylinder block and crankcase assembly	642	6¼
2. Rear assembly	313	6¼
3. Right head assembly	197	12½
4. Left head assembly	197	4¾
5. Reduction gear assembly	88	4¼
6. Injection pump assembly	53	12¼
7. Twelve piston, rod, and bearing assemblies	149	4½
8. Tubing and lines	35	9¼
Total Engine Weight	1677	12

Appendix III

Material Analysis of the Junkers 211B

Part	Brinell Hardness	Fe	Si	Mn	Mg	Al	Cu	C	Cr	Ni	Mo	Va	Others
Injection Plunger	550	Base	0.27	0.38				0.19	0.80	3.7			
Injection Barrel	700	Base	0.27	0.38				0.24	—	—			
Accessory Drive Housing	95	0.12	9.63	0.41	0.19	Base							
Main Bearing Cap	110	0.55	0.79	0.96	1.08	Base	3.99						
Main Bearing Cap Stud	305	Base	0.31	0.57				0.29	2.01	1.91	0.20	0.11	
Camshaft Bearing	40	5.65	0.44	0.08	0.74	Base	0.28						
Camshaft	404	Base	0.21	0.53				0.33	1.51	1.89	0.29		
Camshaft Drive Gear	313	Base		0.61				0.11	2.12	1.83	0.23		
Coolant Pump Housing	60	0.81	11.13	0.34		Base	0.64						0.03 Zinc
Coolant Pump Impeller		0.32	1.71	0.17	4.43	Base							0.03 Zinc
Connecting Rod	362	Base	0.26	0.37				0.31	1.86	2.00	0.30		
Front Crankcase Cover Plate		0.59		0.29	Base	3.64							1.14 Zinc
Crankshaft	311	Base		Trace		Trace	Trace	0.62	0.41	1.63	0.24		
Cylinder Head	95	0.65	10.05	0.03	Base								0.63 Cobalt
Cannon Barrel Idler Gear		Base		0.44				0.20	2.11	1.91	0.22		
Generator Drive Gear		Base		0.39				0.10	0.91	4.49			
Main Reduction Gear	650	Base	0.22	0.53				0.13	1.93	1.85			
Motor Hanger	123	0.37	1.13	0.75	0.40	Base	3.90						
Piston	80-105	0.49	12.22	0.05	0.75	Base	1.06			0.61			
Piston Pin	555	Base	0.29	0.66				0.19	1.85	2.16	0.19		
Piston Pin Plug		0.10		0.38	Base	3.18							1.00 Zinc
Piston Pin Bushing	179	Base	1.40	0.89			0.10	3.25					
Exhaust Valve	210	Base	2.00	0.55				0.48	13.92	12.30			
Exhaust Valve Seat Facing	470	Base	1.8	0.86					28.44				3.1 W, 56.3 Co
Intake Valve	240	Base	1.72	0.30				1.65	14.05			0.30	
Valve Seat Insert	200	Base		0.70				0.50	1.58				
Supercharger Impeller				0.16	Base	6.90							1.34 Zinc
Cylinder Barrel	230	Base	0.28	0.63		0.02		0.44	1.58	0.16	0.04		
Cylinder Stud	320	Base	0.36	0.61				0.29	2.09	2.06	0.18	0.15	
Connecting Rod Bolt	270	Base	0.28	0.57				0.27	1.98	1.78	0.28		
Rocker Arm	330	Base	0.28	0.32				0.31	1.73	1.91	0.40		
Rocker Arm Roller	395&655	Base	0.31	0.56				—	0.70	1.99	0.24		
Crankcase		0.46	9.96	0.47	0.23	Base	0.03		Trace	0.002			0.1 Ti, 0.03 Zn 0.024 Sn 0.005 Pb

FUEL CONSUMPTION from

SECURING accurate data on airline fuel consumptions has been a difficult problem ever since airline transportation came into its own and took its place in the ranks of big business in the United States. Airplane development progressed so rapidly that the airlines hardly had operating procedures well established on one model before a new model came into operation. This succession has continued through the past 20 years, until the Douglas Model DC-3 and DST became the standard airline transport of the United States. Even though the DC-3 is similar in structure for all the airlines, there have been more than five different model engines installed in this basic ship for use in scheduled air transport service in this country. In addition, the DC-3 is being used on long and short schedule operations so that overall fuel consumptions must be studied in relation to the type of schedule as well as to the type of engines and carburetors being used.

When surveying the subject of fuel consumptions in airline operation, consideration must be given to all engine running which requires the consumption of gasoline. For the scope of this paper, we will consider only operation of engines in the airplanes and neglect fuel consumed in overhaul testing of engines.

The purpose of this paper is, then:

1. To break down airline operation into its various phases related to consumption of fuel.
2. To give an idea of the average consumption of fuel for each operating phase.
3. To compare consumptions between the two major methods of mixture control now used on the airlines.
4. To show the possibilities of fuel savings by the airlines during the present emergency.
5. To compare various basic types of fuel as to possible use in airline operation.

In starting a breakdown of fuel consumption in airline operations, it might be well first to look at a scheduled operation Flight Plan as made out by the Captain prior to the flight and also the corresponding Flight Log kept during the flight. Fig. 1 shows the flight plan made by Capt. Hulett for a trip dated Feb. 11, 1942, from Nashville to New York. It is a non-stop flight between the two points as shown by the first two columns.

The symbols in the second column are for the radio range fixes along the route. The mileage between check points is in the third column. The magnetic course for each leg is in the fourth column. The horsepower to be used is in the sixth column. The temperatures anticipated for the altitude to be flown are in the seventh column and are given by the meteorologist on duty at the originating point of flight. The true airspeed in the eighth column is taken from the DC-3 cruising chart. The altitude to be flown is in the ninth column. Wind direction and velocity are in the tenth column. The time between fixes in the last column is calculated from the true airspeed in the eighth column, the heading in the fifth column, and the ground speed in the eleventh column.

At the bottom of the flight plan, the fuel requirements for the flight are made. It is noted that the Captain figured 400 gal were required for a flight of 4 hr, 7 min, or an average of 97 gal per hr. In this average consumption, he allowed a lump sum of 10 gal for taxiing from the loading ramp to the end of the runway, for the run up of each engine prior to take-off, and take-off to 1000 ft altitude. He also averaged the climbing-rich consumption with the cruising-lean consumption to make an average of 97 gal. This actually has a liberal allowance of 3 gal per hr over the average block-to-block consumption of 93.5 gal per hr, calculated from past performance of the whole fleet of DC-3s, when using 600 hp for cruising. The weather map indicated a possibility that ceilings or visibility might go below limits before the trip arrived at LaGuardia Airport, so Hartford was chosen as the closest safe alternate where the weather would be certain.

SECURING accurate data on airline fuel consumptions has been a difficult problem since airplane development progressed so rapidly that the airlines hardly had operating procedures well established on one plane model before a new plane model came into operation. Before greater accuracy can be obtained, several standards must be established. They are:

1. Manufacturers' chart horsepowers based on rpm and manifold pressure are not an accurate basis, at the present time, upon which to base specific consumptions and, until the airlines operate on a torquemeter power basis, accurate comparison between various engine operating phases cannot be made.

2. Specific fuel consumptions must include additional requirements for establishing minimum rich mixture knock ratings in order to insure uniform take-off powers on all fuels purchased throughout the country and to permit airlines to obtain maxi-

The Captain allowed 75 gal for alternate destination and 50 gal for possible instrument approach at Hartford. Instrument approach includes the amount of fuel which might be burned while cruising around awaiting his turn to make the instrument approach. In addition, the Captain added 100 gal reserve which include legal and company reserve, making a total of 625 gal required on board at Nashville to make the flight.

On the reverse side of the Flight Plan is the Flight Log (Fig. 2). The same check points appear on the Flight Log as on the Flight Plan. Time over the fix is recorded in the fourth column, the time over the next check point is estimated ahead, taking into consideration the time made on the last leg and the anticipated wind direction

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the Airline's Viewpoint

by M. G. BEARD

Chief Engineering Pilot, American Airlines, Inc.

for the next leg. The actual time over the fix is recorded in the fifth column and in the seventh column the actual time from check point to check point is listed. The rest of the log is merely a recording of flight as to ground speed, true airspeed, magnetic course, compass headings, wind direction and velocity, as calculated from ground speed and magnetic course, air temperature, altitudes, and horsepower used. A running check is kept on the fuel consumption so that, at any point along the route, a report of consumptions and fuel on board can be made if called for.

At the bottom of the Flight Log is a bracket for fuel consumption. If the passenger and cargo load had been heavy and the reserve fuel was close to the total estimated necessary out of Nashville, a landing might have been made at Washington for fuel, especially if the weather from Washington to New York became worse than the meteorologist anticipated prior to departure of the flight from Nashville. With this particular flight, however, there was no question concerning fuel and consequently the pilot recorded total fuel on board before departure from Nashville and upon arrival at New York. It will be noted that while only 625 gal of fuel were required for the flight, the passenger and cargo load permitted more and the Captain and Flight Superintendent elected to put it on; therefore, 720 gal were put on board at Nashville. Upon arrival at New York, there were 340 gal remaining in the plane, making 380 gal used. The flight lasted for 4 hr, 13 min, making the average hourly consumption 89 gal per hr.

This 89 gal per hr is a round figure and subject to several minor errors which may affect considerably the average figure required on the Flight Log. There are two methods of measuring the fuel in the tanks. It may be measured by reading the meter on the gasoline truck or at the fuel pits while the plane is fueled for the flight. To check this reading, a stick measurement is always made on the quantity in the gas tank after the fueling has been completed. In addition, the plane's fuel-tank gage readings are checked. These three readings may agree, depending upon the accuracy of the fuel truck or pit meters, the accuracy of the fuel gage in the plane, the accuracy with which the stick measurements were made to check this figure, the levelness of the plane as it stands on the ramp, and the care with which the attendant places the stick on the bottom of the tank for measurement.

Securing an accurate estimate of the exact quantity of fuel in a plane, at any instant, whether in flight or on the ground, is one of the most difficult problems related to the study of fuel consumptions. All types of fuel gages thus far developed and now in use in airline operation are subject to errors which, at times, may be as great as 10% of the quantity being measured. Added to gage inaccuracies, is the problem of estimating the amount of residual fuel remaining in the bottom of the DC-3 fuel tank which is, in general, flat but filled with pockets and ridges and crossed by baffles. These irregularities trap between 3 and 10 gal of fuel, depending upon the relative smoothness of the tank bottom and the attitude at which the plane is flying when the tank supposedly runs dry. Future fuel tanks should eliminate one large source of error, and steady progress in the development of accurate fuel quantity gages will greatly assist in obtaining more accurate and reliable data on fuel consumptions in flight; in addition, the trend is now towards transport planes with tricycle landing gear which allows the plane to rest more nearly level on the ground. This feature will eliminate one of the large discrepancies heretofore encountered in the conventional landing gear between tank gage readings on the ground and in flight.

Average fuel consumptions, chargeable against any given airplane, include all ground and air operation. This figure could be divided by the total flying time of the ship to obtain an overall fuel consumption per flying hour, or it could be charged as per day, per month, or per year. This figure is sometimes used in practice for accounting purposes, but the engineer is inclined to regard the overall consumption as too inclusive and to break it down into its component parts, setting aside the less important components and concentrating on the operations which will produce the most returns from careful investigation, experimental work, and continued watchfulness. Block-to-block, or ramp-to-ramp fuel consumptions are very frequently used, especially by the operations department of the airline, as this figure is readily usable directly in calculating fuel requirements for scheduled operation. The block-to-block figure is also too inclusive for the

mum take-off power performance from their engines and with a minimum standard mixture setting for take-off and rated power operation.

3. Instrumentation must be improved for measuring carburetor mixtures, fuel flow in lines, and fuel quantities in fuel tanks in order to increase the accuracy of fuel-consumption data and to allow the airlines to take maximum advantage of established mixture settings and procedures in scheduled operation.

■ ■ ■

THE AUTHOR: M. GOULD BEARD (M '28), after being taught to fly at Kelly Field during World War I, gave up flying following the Armistice to enter the Winton Engine Works where he worked on diesel test stands. Later, after two years as operating diesel engineer in the merchant marine, he joined the diesel engine department of Ingersoll-Rand as experimental and field engineer on marine and locomotive engines. He grew air-minded again in 1927 and joined the Fairchild-Caminez Engine Corp. as installation engineer and test pilot. He is now chief engineering pilot of the American Airlines. Of Yankee missionary parents, he was born in Foochow, China.

engineer as it still contains several miscellaneous operations relatively unimportant to the problem of fuel economy. The phases of engine operation in relation to fuel consumption with which the engineer is most vitally interested, start with the take-off and end when the pilot throttles back and prepares for his initial approach into the field.

Although very little can be gained in economy by a detailed study of the ground operating phases, such a study was made as a matter of general interest and for use in future computations of overall fuel consumptions in airline operation. The various phases of engine operation were broken down as follows:

A. Starting.

B. Engine Warm-Up, Functional Check, and Preflight Check.

C. Taxiing.

D. Engine Run-Up Check prior to Take-Off.

E. Take-Off, including Initial Climb.

F. Cruise Climb.

G. Cruise Level and Descent.

H. Engine Idling.

I. Propeller Dynamic Balancing (Done once or twice per 700 hr period).

A. The fuel consumed in starting an 1830 cu in. engine is almost negligible. The average of a large number of perfect starts is about 0.42 lb per start. Allowing for additional winter priming, it should not exceed 0.5 lb start as a yearly average. The consumption for larger engines is probably proportional to the displacement of the engine.

B. Engine Warm-Up, Functional Check, and Preflight Check – This phase includes the engine cylinder and oil warm-up period after starting the engine, during which the line inspector checks the functioning of the engine and airplane accessories and all instruments; the run up of the engines to check ignition (usually at some prescribed power considerably lower than take-off power); and the run up to check the engines, accessories, and instruments at the terminal points after the flight prior to servicing the plane for the next scheduled run. The time and engine running required for this check average about 15 min for winter and summer, and the fuel consumed is about 50 lb for checking both engines.

C. Taxiing – This operation includes taxiing of planes by line maintenance, taxiing of planes to the loading ramp, and all taxiing to and from the terminal in schedule operation. The amount of taxiing varies greatly with the size of the field. Studies made at Chicago in 1938 allowed an average of 2 min ground operation on landing and 2 min for take-off including all runways. The power used seldom exceeds bursts of 300 hp and probably averages about

AMERICAN AIRLINES, INC.
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FLIGHT PLAN

CAPT. HULETT
1ST OFF. NAGE

FROM NA TRIP 23-4 DATE 2-11-42
TO NY AIRPLANE

FROM	TO	MILEAGE	MAGNETIC COURSE	COMPASS HEADING (UNCORRECTED FOR DEVIATION)	HORSE POWER	TEMP.	TRUE AIRSPEED	ALT.	WIND DIRECTION & VEL.		GR'ND SPEED	EST. TIME CP.-CP.*	EST. TIME TOTAL
NA	XM	51	98	94	600	30	~~120~~ 185	9000	300	25	153	:20	:
	KX	102	94	91		30	185	9000	300	25	208	:29	:
	JB	103	66	62		25	185	9000	280	25	206	:30	:
	US	102	66	62		25	185	9000	280	25	206	:30	:
	RO	45	73	70		20	185	9000	270	30	212	:13	:
	GJ	112	65	61		20	185	9000	270	30	212	:32	:
	FMS	67	66	62		20	185	9000	270	30	212	:19	:
	WA	21	18	10		20	185	9000	270	30	192	:06	:
	XBT	35	47	43		20	185	9000	270	20	200	:11	:
	PG	37-50	47-80	43 79		25	181	7000	270	20	201	:26	:
	XLI	84	73	71		25	170	1500	270	15	197	:25	:
	LG	14	44	42		30						:06	:
												:	4:07

REFUELING POINTS	FROM	TO	FROM	TO	FROM	TO	FROM	TO	FROM	TO
	NA	NY								
ESTIMATE GALLONS GAS: REFUELING POINT TO REFUELING POINT	400									
DESTINATION TO ALTERNATE	75									
INSTRUMENT APPROACHES & ATC.	50									
TOTAL INCLUDING RESERVE	625									
ALTERNATE AIRPORTS	HT									

STATE DESTINATION IN CASE OF TOTAL RADIO FAILURE WA-

*—CHECK POINT TO CHECK POINT

CAPTAIN

FLIGHT SUP'T ________ METEOROLOGIST

THIS FORM TO BE MADE OUT IN TRIPLICATE.

Fig. 1

110 hp for a twin-engined transport of 25,000 lb gross weight. Checks made at LaGuardia Field show an average of 3 lb per minute fuel consumed during normal taxiing conditions for the DC-3.

D. Engine Run Up Check prior to Take-Off – This operation is required as a final check of all functioning units of the powerplant to ascertain that everything is in order prior to the take-off. Throttles are advanced individually to a specified manifold pressure and held until the ignition is checked on both magnetos and the engine rpm, oil pressure, fuel pressure, and so on, are checked. This usually requires about 8 to 12 sec for each engine. The total consumption for an 1100-hp engine run up at 30 in. manifold pressure at about 2000 rpm for 10 sec is 2 lb.

E. Take-off, including initial climb – The airlines vary considerably in use of full rated take-off power on the DC-3. Most of the airlines operating engines which develop take-off power at sea level are using full allowable power on each take-off. Other airlines use 85% of rated take-off power except where more is needed. The take-off power used is held until the plane is clear of all obstructions adjacent to the field or until the plane is about 50 ft in the air. This time seldom exceeds 45 sec and normally is about 30 sec for land planes. Power is reduced to climbing power in steps; first, by retarding the throttles to approximately rated power position; then reducing the rpm by moving the propeller controls to climbing rpm position; and finally, adjusting the throttles for climbing power. The complete operation from start of take-off requires, on an average, about 60 sec.

The mixture setting of the carburetors for take-off power varies considerably between airlines operating the same type of engines on the same type of fuel, and with engines having the same type of cowling. Surveys indicate that take-off power mixtures vary between 0.72 and 0.81 lb per bhp-hr which corresponds to 0.106 to 0.120 F/A. A very interesting fact is that some installations equipped with cowl flaps are using take-off mixture settings richer than some installations with a fixed gap cowl. All operators are losing an appreciable amount of power on take-off because of the rich mixtures used. Fig. 3 is a curve plotted from torquemeter tests in flight to determine the relation of power to mixture. This curve was not carried beyond 0.114 F/A as the carburetor used would not go beyond this mixture. All engine manufacturers can give accurate data obtained on dynamometer tests so that the operator should be aware of the approximate penalty on take-off power incurred by the use of any given mixture. It might be added here that, prior to the use of torquemeters for airplane performance testing, results

FLIGHT LOG

FROM	TO	TIME OFF	ESTIMATED TIME OVER NEXT CHECK	TIME OVER FIX OR CHECK	MINUTES CHECK POINT TO CHECK POINT: ESTIMATE	ACTUAL	MILES FROM LAST CHECK	GROUND SPEED	TRUE AIR SPEED	MAGNETIC COURSE	COMPASS HEADING (CORRECTED FOR DEVIATION)	WIND DIRECTION AND VELOCITY	TEMP.	ALT. FLOWN	HORSE POWER USED
NA	XM	:08	:28	:29	:20	:21	51	145		98	95		30	9000	600
	KX	:	:58	:59	:29	:30	102	204	183	94	90	310 - 25	29	9000	600
	JB	:	:29	:29	:30	:30	103	246	183	66	64	260 - 25	30	9000	600
	US	:	:59	:58	:30	:29	102	212	183	66	64	255 - 30	30	9,000	600
	RO	:	:11	:10	:13	:12	45	225	183	73	69	250 - 35	30	9,000	600
	GJ	:	:41	:42	:32	:32	112	210	183	65	60	260 - 30	28	9000	600
	FMS	:	:01	:01	:19	:19	67	210	183	66	60	260 - 30	25	9000	600
	WA	:	:07	:07	:06	:06	21	210	183	18	16	260 - 30	24	9000	600
	XBT	:	:18	:17	:11	:10	35	210	183	47	43	250 - 30	24	9000	600
	PG	:	:43	:40	:26	:23	87	227		47-80	47-78	-	28	7000	600
	XLI	:	:03	:07	:25	:27	84	186		73	70	-	30	1500	600
	LG	:	:09	:13	:06	:07	14	120		44	40	-	34		600
		:	:	:	:	:						-			
		:	:	:	:	:						-			
		:	:	:	:	:						-			
		:	:	:	:	:						-			
		:	:	:	:	:						-			
		:	:	:	:	:						-			
		:	:	:	:	:						-			
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		:	:	:	:	:						-			
		:	:	:	:	:						-			
		:	:	:	:	:						-			

FUEL CONSUMPTION

		STA.	GALLONS	STA.	GALLONS	STA.	GALLONS	STA.	GALLONS	STA.	GALLONS
FUEL IN TANKS	ON DEPARTURE FROM FUELING STATION	NA	720								
	ON ARRIVAL AT FUELING STATION		340								
TOTAL GALLONS USED		380									
FLIGHT TIME		4:13									
AVERAGE GALLONS PER HOUR		89									

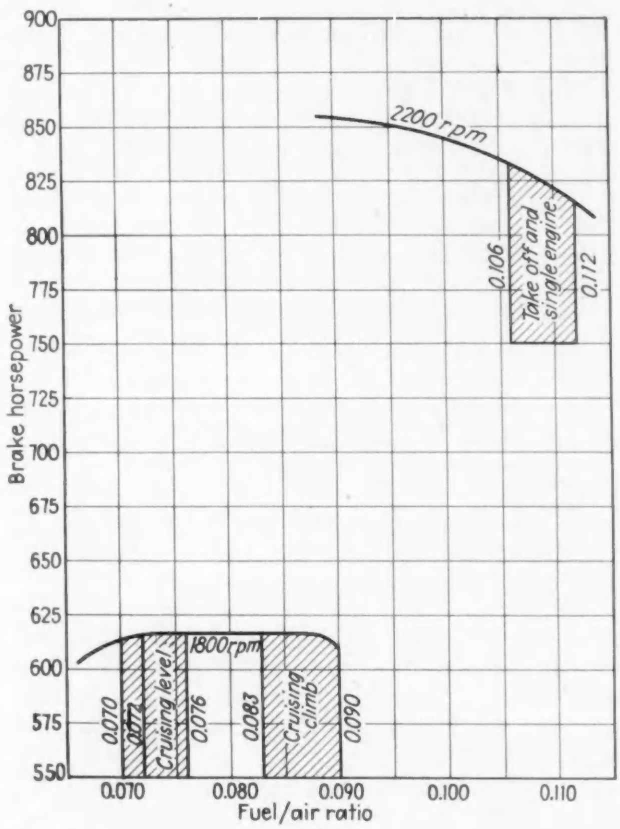
REMARKS

0 STATUTE 10 MILES 20 30 40 50 60 70 (SCALE FOR 38° LATITUDE)

■ Fig. 2

in take-off and rated power performance were very erratic and flight analysts were under the necessity of using an average of a large number of tests in order to obtain a reasonably accurate result. The use of torquemeters, analyzers, and flowmeters made it possible to duplicate (with reasonable accuracy) any test made at an earlier date. On account of overheating with lean mixtures and power loss with rich mixtures, it is necessary to control carefully the mixtures of all carburetors in operation in order to obtain uniform take-off performance from all planes in the fleet.

Because the power varies considerably from the start



■ Fig. 3 – Horsepower change with mixture for single engine and cruising powers – taken from torquemeter tests made at Douglas factory December, 1939, on DC-3, NX-21749

of the take-off until climbing power has been established, the whole sequence has been treated as one operation in determining the total amount of fuel consumed during the take-off operation. An average of several tests on a DC-3 equipped with two 1100-hp engines having carburetors set at 0.112 F/A, shows that the average take-off consumes 25.5 lb or 4.2 gal of fuel. In estimating the required fuel capacities, operations departments of some airlines use a round figure of 10 gal for all ground operations and take-off. From the foregoing analysis, points A, C, D and E total 36 lb or 6 gal. It would appear that 10 gal is allowing a very adequate margin for reserve and for additional ground holding in all weather where engines may be idled for as much as 40 min prior to take-off. Under fair weather conditions, the payload of the ship could be stepped up about 24 lb if this round figure of 10 gal were reduced to 6 gal total consumption between the time the tanks are topped off prior to flight until the power is reduced for cruising climb after take-off.

F. Cruise Climb – Most of the airlines allow more power for cruising climb than for cruising level operation as this phase of the operation extends over a shorter time and it is desired to get the plane to cruising altitude in the shortest reasonable time. Allowable climbing power varies from 65% to 70% of METO power among the airlines. Climbing airspeeds used vary between 110 and 125 mph, but normally are about 65% of the cruising speed. This reduced speed decreases the cooling blast for the engine and consequently requires additional cooling by use of rich mixtures. A survey of the airlines shows that climbing mixtures vary between 0.54 and 0.59 lb per bhp-hr which corresponds to 0.082 to 0.090 F/A. In general, it appears that users of cowl flaps can use slightly leaner mixtures for climbing operation than can the users of installations with fixed gap cowls.

On routes where the average flight is about 45 min, block to block, and where cruising altitudes are relatively low, the cruising-climb operation offers little opportunity for fuel savings, as frequently the climb is completed within a few minutes after take-off. However, on the longer schedules where the climb continues for 15 to 20 min, climbing consumptions offer good possibilities for reducing yearly consumptions of fuel. Looking further into the future toward planes which will fly at altitudes above 20,000 ft, climbing consumptions will offer even greater opportunities for improving the general fuel economy of the airline.

Climbing mixtures are always based on the requirements of the hottest cylinder. Cylinder-head temperature distributions nearly always vary considerably from head temperature distribution in cruising level flight. Usually, the top cylinders of a radial aircooled engine are hottest in climbing operation. If suitable cowling can be obtained to effect better cooling of the top cylinders in climb, it will be possible to lean climbing mixtures and effect a substantial saving in fuel consumption during climb.

G. Cruise Level and Descent – Since cruising level flight and the descent are usually made at the same power, they will be dealt with under the same heading. These phases of the operation offer the greatest possibilities for an overall saving in fuel consumptions. Cruising level and descent comprise from 60% to 80% of all airline engine operation and consequently, more time and effort have been expended on obtaining economical consumptions for this phase of engine operation than for any of the others heretofore mentioned.

H. Engine Idling – Engine idling consumptions in lb per min prove to be so small in quantity that slight allowances have been made for this quantity in the taxiing consumptions and engine run-up consumptions. The one case where engine idling amounts to an appreciable figure is the infrequent occurrence where a plane is held for sometimes as much as 45 min in thick weather, awaiting its turn to take off from the airport of departure. The airlines usually allow about 10 gal extra in the tanks for departure when this condition is anticipated.

I. Propeller Dynamic Balancing – Another item of fuel consumption that comes into the overall picture is that consumed in dynamic balancing of propellers. This operation is made only on engine and propeller installations with 3:2, 16:9, and 2:1 gear ratios. The operation is conducted once or twice in the engine overhaul period and

consumes on an average about 25 gal for balancing both propellers on the DC-3.

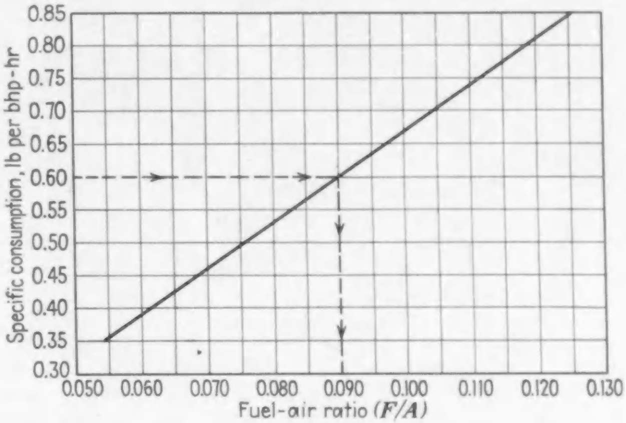
There are two methods being used at the present time by the airlines for adjusting carburetor mixtures in flight. The first is by manual adjustment of mixtures to the desired fuel-air ratio suitable for the power being used and the airspeed of the particular flight. The second method is by automatic mixture adjustment by use of a load, temperature, and altitude compensating carburetor. This first method has the advantage of being sufficiently flexible so that the pilot can adjust the mixtures to the particular conditions of flight and to the needs of the engine for cooling as regards cylinder head temperatures. This method requires some attention by the pilots to adjust the mixtures for the varying conditions of flight in order to obtain the desired minimum fuel consumption. With two pilots in the cockpit of a twin-engine transport, this method has worked out very well and has resulted in a reasonable average yearly fuel consumption. Prior to the use of constant-speed propellers, the fixed-pitch or two-position propeller gave a rough indication for mixture setting by carefully noting the change in rpm in steady flight at any given attitude of the airplane as the mixture was leaned out from full rich and also by keeping cylinder heads close to desired temperatures. When constant-speed propellers came into use, this method was no longer usable and the Cambridge analyzer was adopted for an indicator of carburetor mixture settings.

The Cambridge analyzer became a reliable instrument as soon as installation difficulties had been ironed out. There are still some temperature errors which, at the present time, are being averaged but which may be eliminated in the future by a temperature-compensated sampler unit. The present analyzer in use cannot be relied upon for accurate readings without frequent calibrations for mixtures much below 0.070 F/A. Therefore, if specific consumptions below 0.45 are desired in airline operation, another form of mixture control is desirable. Engines and installations designed prior to 1938 usually have certain limitations in their ability to operate at mixtures below 0.070 F/A at 70% power in transport airplanes whose cruising speed is less than 185 mph. Therefore, the exhaust gas analyzer still provides a convenient and reliable method for adjustment of the desired mixtures. Fig. 4 shows the general relation between specific consumptions and fuel air ratios as calculated from a large number of torquemeter flight tests in a DC-3.

The efficiency of adjusting mixtures manually and depending upon pilots to adjust the mixtures, depends upon the alertness of the crew in changing the mixtures as flight conditions change and in their care in securing accurate adjustment of the mixtures, and in regular and careful servicing of the exhaust-gas analyzer. With this system, carburetor basic settings are usually a little richer than desired except for take-off power where the mixture is set for minimum desired in order to decrease the power loss with mixture controls in full-rich positions. When both members of the crew are busy during bad weather with their navigation and radio contacts, mixture adjustments are inclined to wait until a suitable time before adjustments are made. If hurried, the pilot is inclined to leave the mixtures richer than necessary during climb in order to protect the engines from damage under conditions when he does not have time to attend to the adjustment with precision.

Flowmeters indicating directly in pounds of fuel consumed per hour have been developed within the last two years which give reasonable accuracy. Fuel totalizers have been used for several years. These devices, however, are impractical for use in transport planes which do not carry flight engineers and where there are only two members in the flight crew, as it requires too much time and calculation to adjust mixtures by their use.

Remote indicating flowmeters, reading directly in pounds per hour, have been used successfully for manual adjustment of mixtures on one airline also using the exhaust-gas analyzer for the same purpose. Average fuel consumptions



■ **Fig. 4 – Correlation between specific fuel consumption and fuel-air ratio – taken from tests on DC-3 equipped with G-102 engines, torquemeters, flowmeters, and Cambridge analyzers; also from averaging Douglas and American Airlines curves from DC-3 tests and Wright Aeronautical Corp. data taken on test stands**

over a year's operation on four DC-3s equipped with flowmeters show that these planes have about the same average block-to-block fuel consumption as the planes equipped with exhaust-gas analyzers. The flowmeter indicates immediately any change in the quantity of fuel being delivered to the engine and therefore the desired mixture setting is more quickly attained than by the exhaust-gas analyzer. Also, a direct figure for consumption and weight reduction is indicated which the pilot can use without additional calculations.

In adjusting mixtures by the flowmeter, the horsepower must first be established by adjusting to the desired rpm and manifold pressure. As soon as the horsepower is adjusted, the desired mixture can be adjusted without delay. When using the exhaust-gas analyzer for mixture adjustment, there is a delay required of from one to two minutes before the analyzer indication stabilizes. The flowmeter has the additional advantage in that it will indicate fuel flow regardless of the leanness of the mixture. One disadvantage of manually adjusting mixture by the flowmeter is that, in hot weather, the fuel lines are filled with a column of fuel mixed with a considerable volume of air bubbles. This, in itself, creates a considerable error in fuel meters of the positive-displacement type. Flowmeters with the vein-type mechanisms are only slightly affected by air bubbles in the fuel. Large air bubbles create considerable fluctuation in the fuel flow, especially if the carburetor is equipped with an air separator, since the unloading of the air separator allows a sudden gush of fuel to fill up the space emptied of air. This pulsating makes the flowmeter indicator fluctuate accordingly and makes it difficult to attain an accurate adjustment of

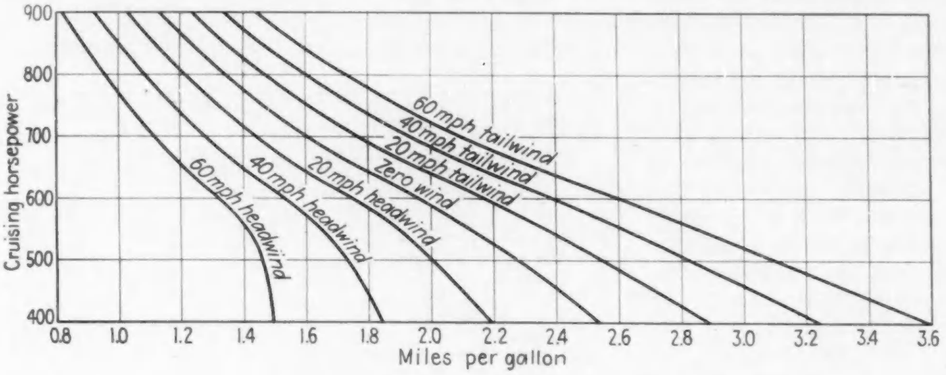
mixtures. Averaging the fluctuation of a flowmeter is rather difficult as the pulsations are erratic and far from regular, and it is difficult to average the reading in relation to the time the needle remains at one or the other end of the fluctuation.

The automatic carburetor is an improvement in the technique of controlling mixtures in scheduled operation, especially in the twin-engine transport used on schedules where frequent stops are made and average time between stops is less than 1.5 hr. It has resulted in slightly lower overall fuel consumptions. Although the principles of operation of the two automatic carburetors now being used are different, the results achieved are substantially the same. There is a set position of the controls for high power operations such as take-off and single-engine operation; another position for climbing and cruising rich operation; and a third position for cruising lean operation. Almost no provision is made for intermediate mixtures between cruising rich and cruising lean. It is possible to obtain some intermediate adjustment, but the control is very sensitive and not practical for normal operation.

With the automatic carburetor, the pilot has only two positions for climb and cruise operation to worry about, and the probabilities are that the mixtures will be more closely adjusted according to the flight operation involved than with the manual method of mixture adjustment.

For airline operation, where the engine operating conditions are uniform, this system works satisfactorily. Under some conditions, it is necessary to operate richer than is necessary on account of the two-position mixture control but, as the percentage of time over which the rich mixture is required is a very small proportion of the total operating time, the loss in fuel economy is very small. Once the correct cruising lean mixture has been established which is suitable for all cruising level flight conditions, all carburetors can be adjusted accordingly and cruising consumptions can be held uniformly on all airplanes of the fleet. The greatest savings under this condition are probably brought about by taking the ideas of the individual pilot concerning the correct mixture at which to operate out of the picture and allowing him only one mixture for cruising lean operation. Many pilots are inclined to operate on the rich side of any recommended mixture setting under the manual mixture adjustment method. With an automatic carburetor, the mixture is pre-set for the pilot so that, instead of controlling the mixture to an infinite degree, he uses cruising lean position unless some condition of the engine requires a richer mixture, at which time he goes to cruising rich. The percentage of times at which this emergency arises is relatively small, and the net result is a saving in fuel consumption during cruising and climbing operation over the manual adjustment method.

Although the automatic carburetor has proved its value on the type of operation in which the DC-3 is now used, it is not the most economical method of mixture adjustment for a long-range operation such as is being conducted on transoceanic routes and will be conducted in long-range transcontinental operation for four-engined transports after the war. Since it is necessary to establish the cruising-lean mixture of an automatic carburetor at a specific consumption that will be rich enough to cover a wide range of cruising-lean conditions, it is impossible to secure the leanest mixture possible for the changing powers and airspeeds most economical for long-range cruising operation. There is still considerable controversy as to whether four-engined land planes require more than two men in the cockpit crew, especially if most of the operating accessories are made automatic in operation. Providing a suitable system of manual adjustment can reduce the average cruising consumptions by 0.03 lb per bhp-hr under that of automatic adjustment, a flight engineer on a four-engined transport using an average of 900 hp per engine for cruising can save his salary and payload weight in about 2 hr flying per trip.



■ Fig. 5 – Miles per gallon versus cruising horsepower – Douglas DC-3 airplane assumed cruising at 7000 ft – gross weight 24,800 lb

Fig. 5 shows the cruising economy of a DC-3 in miles per gallon versus horsepower at 7000 feet average cruising altitude and at 24,800 lb gross weight. The mixture is assumed to be 0.071 F/A or 0.465 lb per bhp-hr. Consumption lines for winds of $\pm$ 20, 40, and 60 mph are shown either side of the zero wind line. From the following tabulation, it is noted that, for an average 300-mile cruising flight, the fuel saving of 12 gal can be effected by a reduction of 50 hp with an increase in cruising time of 5 min. Fuel saving of about 21 gal is effected by the reduction of 100 cruising hp with an increase in time of 10 min.

Hp	Mpg	True Air Speed Cruising, Mph	Time for 300 Miles	Time Increase, min	Fuel Consumption, gal	Fuel Saving, gal
600	1.96	183	1:38	—	153	—
550	2.13	175	1:43	5	141	12
500	2.28	167	1:48	10	132	21

A transcontinental schedule of approximately 15 hr flying time would be increased about 36 to 40 min in time by a reduction of 50 cruising hp and 1 hr and 20 min by a reduction of 100 cruising hp from the 600 hp on which the average DC-3 transcontinental schedule is now operat-

ing. Considering that the total mileage flown by all airlines operating DC-3s in this country during 1941 was approximately 118,500,000 miles, of which about 85% or 104,000,000 miles was cruising level and descent flying, the fuel consumption at 600 hp with an average of 10-mph headwind condition, would be roughly 56,250,000 gallons for the cruising operation alone (neglecting take-off and climb consumptions). If schedules were set up on a 550-hp basis, the analagous cruising consumption would be 51,900,000 gal or a saving of 4,350,000 gal per annum. If cruising powers were further reduced to 500 hp, the analagous consumption would be 48,400,000 gal with a consequent saving of 7,850,000 gal per annum from the consumption at 600 cruising hp. This represents approximately 10% of the total consumption on the airlines for 1941. While this figure is not large when compared to the anticipated military and commercial consumptions for 1942-1943, it would be a substantial saving in the event of a critical situation in aviation gasoline.

Aviation fuel specifications thus far used have taken into consideration the lead content, the octane rating, evaporating characteristics, gravity, and heat content of the fuel. About three years ago, fuel experts became aware that fuels which met the general specification differed greatly in the power they would produce under rich-mixture conditions at high powers. Fuel samples from various parts of the country, all of which met the general fuel specification for airline operation, have been tested by both large engine manufacturers with the result that maximum rated take-off horsepower varied as much as 11% at the point of detonation. Special fuels with high aromatic content have been blended which will produce even a greater spread in take-off power at any given rich mixture. The performance of an airplane can be considerably affected by such a power variation on take-off. The foregoing figures would vary the take-off power of two 1200-hp engines in a DC-3 as much as 260 hp. A four-engine transport with 8000 hp installed could be subject to as much as 880 hp variation when operated on different fuels, all of which would meet present fuel specifications. For the future fuel specifications, after the present emergency, the octane number must be pegged by an additional specification which will provide for a minimum-rich-mixture knock rating as well as a minimum-lean-mixture knock rating which is done in the present practice. Such a specification should guarantee fuels of uniform rich mixture power characteristics.

In the past, it has been impossible to make an accurate estimate of specific consumptions used by the various airlines for various types of operations because of the lack of uniformity in keeping records of fuel consumptions and because of the different basis upon which each airline computes its fuel consumption. The discrepancy is increased because of the lack of accurate power measurement. Tests on DC-3s conducted with torquemeters, flowmeters, and exhaust-gas analyzers, indicate that there is a discrepancy averaging about 4% between engine manufacturers chart powers and actual torquemeter power delivered to the engine. As an example a certain test conducted at 600 chart hp, showed a consumption of 252 lb of fuel per hour, making the specific consumption 0.42 lb per bhp-hr. The torquemeters, however, indicated that 575 hp was being delivered to the propeller. On a torquemeter power basis, the specific consumption was 0.437 lb per bhp-hr. Such discrepancies existed throughout the tests, being a minimum when the engines were operated at best mixture settings and being a maximum when the mixtures varied farthest from best settings. Referring to Fig. 3, the lower curve shows the effect of lean mixtures on engine torquemeter horsepower. On this particular engine, the torquemeter power was fairly uniform from best setting down to 0.074 F/A but, as the mixture was reduced below .074 F/A, the power fell off rapidly until at 0.068 F/A, it was approximately 12 hp lower than at best setting.

When using engine chart powers on which to base engine power settings, there is no way by which the pilot can accurately determine the power being delivered to the propeller. Indicated airspeeds may be used as a rough approximation in relation to a chart showing speed power relations based on torquemeter tests. Because of rising and falling air currents, airplane load conditions, and so on, this method may introduce even greater errors into the picture. The only adequate method thus far found is the use of torquemeters on the engines for use in power adjustment. At the present time, torquemeters are being used only in very special cases for power settings. Engines designed up to within a year ago have made no provision for the inclusion of torquemeters in the engine nose case. Some manufacturers, however, are recognizing the necessity and the possibilities for better engine operation by the use of torquemeters and it is hoped that future transport planes will be equipped with engines incorporating torquemeters for power adjustment. When torquemeters are used for airline operation, operators will be able to effect a closer check on their operating fuel consumptions.

■ Summary

In summation, it is evident that comparisons of fuel consumption in airline operation at the present time cannot be made too accurately. Before greater accuracy can be obtained, several standards must be established:

1. Manufacturers' chart horsepowers based on rpm and manifold pressure are not an accurate basis on which to establish engine power. Torquemeter power is the most accurate basis, at the present time, upon which to base specific consumptions and, until the airlines operate on torquemeter power basis, accurate comparison between various engine operating phases cannot be made.

2. Specific fuel consumptions must include additional requirements for establishing minimum-rich-mixture knock ratings in order to insure uniform take-off powers on all fuels purchased throughout the country and to permit airlines to obtain maximum take-off power performance from their engines and with a minimum standard mixture setting for take-off and rated power operation.

3. Instrumentation must be improved for measuring carburetor mixtures, fuel flow in lines, and fuel quantities in fuel tanks in order to increase the accuracy of fuel consumption data and to allow the airlines to take maximum advantage of established mixture settings and procedures in scheduled operation.

■ Acknowledgment

The writer wishes to express appreciation for aid and assistance in compiling information for this paper to the Research and Engineering Departments of Braniff Airways, Inc.; Chicago & Southern Air Lines, Inc.; Eastern Air Lines, Inc.; Pennsylvania-Central Airlines Corp.; Pratt & Whitney Aircraft; Transcontinental & Western Air, Inc.; United Air Lines Transport Corp.; Wright Aeronautical Corp.; and to his associates in American Airlines, Inc.

Engineering Problems of

by R. D. KELLY and W. W. DAVIES

United Air Lines Transport Corp.

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IN the enormous increase in air-cargo transportation predicted for the near future, good schedule reliability is an absolute need. If schedule reliability is not maintained, it will be impossible to make close connections with other air carriers and time thus wasted would overcome many of the advantages of shipment by air.

To attain such schedule reliability, cargo airplanes must be equipped with all the facilities needed for operation under adverse conditions, including anti-icing equipment and a full complement of radio and other necessary instruments.

The purpose of this paper is to indicate probable design and performance trends and to outline some of the specific problems of carrying air cargo. The conclusions drawn are based upon an actual research study of cargo airplane performance and design criteria, and upon present airline cargo handling experience. These conclusions indicate that present methods and equipment are wholly inadequate for any great extension.

Cargo airplanes, making frequent stops, provide a logical means for transportation of passengers for short distances, or to the major cities where they can be transferred to through flights.

The authors discuss the actual engineering and mechanical problems involved in the handling of air cargo under the following headings: performance cost and size consideration; design for efficient handling of cargo within the airplane; and equipment outside the airplane.

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THE AUTHORS: R. D. KELLY (M '29), superintendent of development with United Air Lines Transport Corp., decided upon aviation as a career when a barnstormer gave him his first plane ride upon completion of his sophomore year at Franklin College, Indiana, where he was majoring in chemistry. Pursuing his aviation bent, Mr. Kelly later joined Boeing Air Transport and was placed in charge of the instrument shop at their Cheyenne base. In 1936, after this company had become a part of United Air Lines, he went to Chicago as test engineer, and in 1937 was put in charge of the company's Engineering Development Group. During the past year, Mr. Kelly has been engaged primarily in research relating to the economic aspects of future airplane designs. W. W. DAVIES (J '40), assistant superintendent of research, United Air Lines Transport Corp., has conducted numerous experiments in high-altitude flying in its relation to the physical well-being and comfort of passengers and crews. When a new world's altitude record for transport planes was made in a United Air Lines *Mainliner* ascending 29,800 ft with a group of University of Chicago scientists to photograph cosmic rays, Mr. Davies was in charge of the oxygen equipment aboard the plane. He is a graduate of the Armour Institute of Technology.

AN enormous increase in air cargo in the near future has been predicted by various authorities, and many surveys have been made to analyze the markets which should develop. But, to the knowledge of the authors, little has been said about the actual engineering and mechanical problems involved in the handling of air cargo. It is true that several cargo plane designs have been evolved and proposed which include many novel construction features, but little has been available to designers in the way of operation and cargo handling recommendations.

It will be the purpose of this paper to indicate probable design and performance trends and outline some of the specific problems of carrying air cargo. The background for the conclusions drawn is based upon: first, an actual research study for cargo airplane performance and design criteria; second, present airline cargo-handling experience, from which it is apparent that present methods and equipment are wholly inadequate for any great extension; and third, opinion and imagination.

At the very first of this paper it is desired to emphasize one very important fact. An airplane only progresses towards its destination when it is in flight. Therefore, all ground time during scheduled operation is lost time, and every effort must be made to keep this to an absolute minimum. For instance, on the existing non-stop schedule from Chicago to New York, every minute of delay in getting into the air must be compensated for by an increase of 1 mph in flight speed. Converted into horsepower, this means 11 hp more for the entire trip in a DC-3 airplane for each minute's delay.

Using this same type of illustration for a transcontinental cargo trip, suppose we assume a scheduled 2600 miles in 13 hr, or an average speed of 200 mph. Let us also assume eight stops enroute, which is certainly not excessive for a cargo flight. If an average of 20 min ground time is required for each stop, then the average flight speed the airplane must make good to maintain schedule will be:

$$\frac{2600}{13 - \frac{(8 \times 20)}{60}} = 252 \text{ mph.}$$

If the ground time can be cut in half, to 10 min, then the equation becomes:

$$\frac{2600}{13 - \frac{(8 \times 10)}{60}} = 223 \text{ mph.}$$

This demonstrates a difference of 29 mph. Where else

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AIR CARGO TRANSPORTATION

can an effective improvement of 29 mph be gained so easily or cheaply as by improving ground-handling methods and by efficient cargo storage in the airplane?

Probably many designers and operators think that even a 20-min average ground time per stop for fueling and cargo transfer is fairly good – and so it is, for present thinking and methods. But, are they willing to sacrifice so much speed and economy to a competitor who has the equipment and methods to get his ship in and out in 10 min? Every minute gained in reducing ground time helps throughout the whole trip. Further, the reduction in congestion due to fewer airplanes being on the ground at one time is of major importance also.

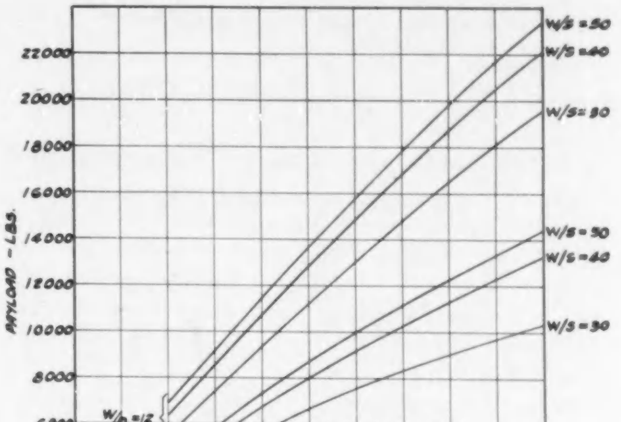
The railroads realize the importance of short stops. Recently one of the writers had an opportunity to ride in the cab of one of the latest streamliners. The major surprise came at a division point when, even with a crew change, the time between the stop and start was less than 2 min. Passengers, mail, express, and baggage were all easily handled within this time.

Another important point to be stressed is the absolute need for good schedule reliability for air-cargo operation. Many proposals for cargo airplanes have made the assumption that air cargo could wait for favorable weather, and that schedule reliability and safety were not of primary importance. Further, that vibration, noise, lack of heating and ventilation, and so on, could be tolerated if weight and cost of the airplane could be reduced thereby. It is the opinion of the writers that this attitude is incorrect, and that future designs should be based upon a schedule reliability, safety, and general operation closely approximating that of the purely passenger transport operation. Most of the reasons for this thinking will be developed later in the paper but, at this time, it may be well to bring out one or two important factors.

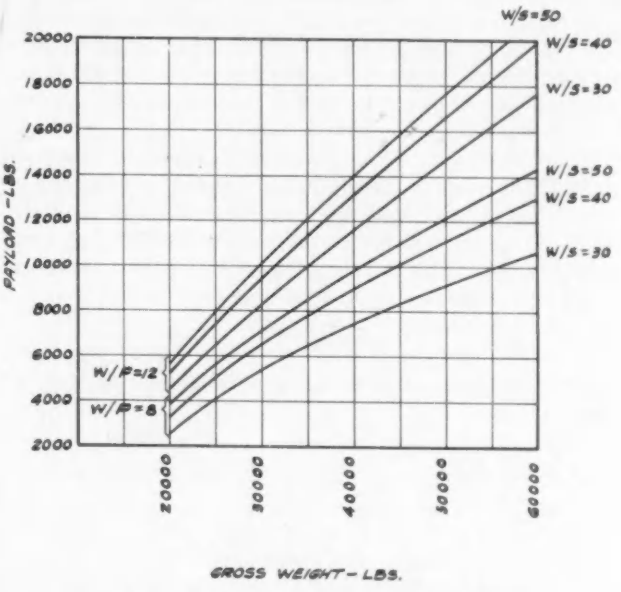
If schedule reliability is not maintained, it will be impossible to make close connections with other air carriers, and time thus wasted will overcome many of the advantages of shipment by air. In almost every case where material is sent by air, a premium is paid for that service because time is important and a certain deadline is to be made. Without a schedule reliability as good or better than that of the passenger transport, the customer is gambling upon the arrival time of his shipment at its destination. This does not promote the maximum use of air cargo.

To attain schedule reliability it is necessary that cargo airplanes be equipped with all the facilities needed for operation under adverse conditions. They must be provided with anti-icing equipment and a full complement of radio and instruments. The fact that a greater number of stops will be made on a given route by the air cargo airplane will mean that it will usually need to be operated with greater efficiency and precision than in the case of the through trip whose primary function will be to carry passengers and mail long distances with few stops.

Cargo airplanes, making frequent stops, provide a logical means for transportation of passengers for short distances, or to the major cities where they can be transferred to through flights.



■ Fig. 1 – Payload trend – 200-mile range



■ Fig. 2 – Payload trend – 500-mile range

Table 1 – General Characteristics

Gross Weight, lb	20,000			30,000			40,000			50,000			60,000		
No. of Engines	2	2	2	2	2	2	3	2	2	4	3	3	4	3	3
METO Hp/Engine	1250	1000	835	1875	1500	1250	1660	2000	1660	1560	1660	1390	1875	2000	1660
Aspect Ratio	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10
Fuel Capacity, gal	800	700	600	1250	1000	800	1550	1350	1200	1950	1800	1550	2300	2050	1900
Crew	2	2	2	2	2	2	3	2	2	3	3	3	3	3	3

NACA laminar-flow wings.
Full insulation for cabin and cockpit, and as required for cargo compartments.
Integral fuel tanks.
Nose wheel.

A. Performance Cost and Size Considerations

Basically the performance and general external appearance of a cargo airplane will not differ from any similar-sized passenger transport. Both types of airplanes must do the same job, that is, carry a given load over a given distance in a certain length of time. The primary requirements for any airplane are defined by speed, payload, and trip length or lengths.

The payload requirements of the cargo airplane are defined very definitely by the existing cargo market, which will establish the size of the plane depending upon the frequency of schedule. Schedules may have to be operated frequently to suit the demand, thus calling for a greater number of small airplanes or, when the demand will permit the holding over of certain cargo, a smaller number of large airplanes may be used.

It is generally accepted that speed is of major importance for transportation of passengers. It is the opinion of the authors, and several cargo and express surveys agree, that speed is also of prime importance to the shipper. Block-to-block time for a cargo plane must be kept comparable to that of the passenger plane. Cruising velocity of cargo planes must then be increased if the cargo airplane is to make more frequent stops.

Trip length may differ between a passenger transport and a cargo transport because of the importance of getting passengers through with the minimum number of interruptions. Cargo will not be affected by frequent stops as long as speed is not greatly impaired. It appears that trip lengths of cargo airplanes in the immediate future can be expected to be from 150 to 700 miles.

It is believed that a large transport system could be properly equipped to perform its function by using equipment as follows:

1. *Long-haul* deluxe *passenger airplane* having minimum cargo provisions to handle only luggage and possibly mail.

2. *Medium-* and *short-haul passenger plane* with *moderate* cargo provisions. This cargo space would be located approximately the same as at present, but having a more efficient design and greater volume. Medium-haul passenger loads and light cargo loads justify this unit.

3. *Medium-* and *short-haul cargo plane* with *moderate passenger* provisions. This airplane functions primarily as a major cargo operating unit, but also acts as a pickup for passengers to transfer to the through passenger operation.

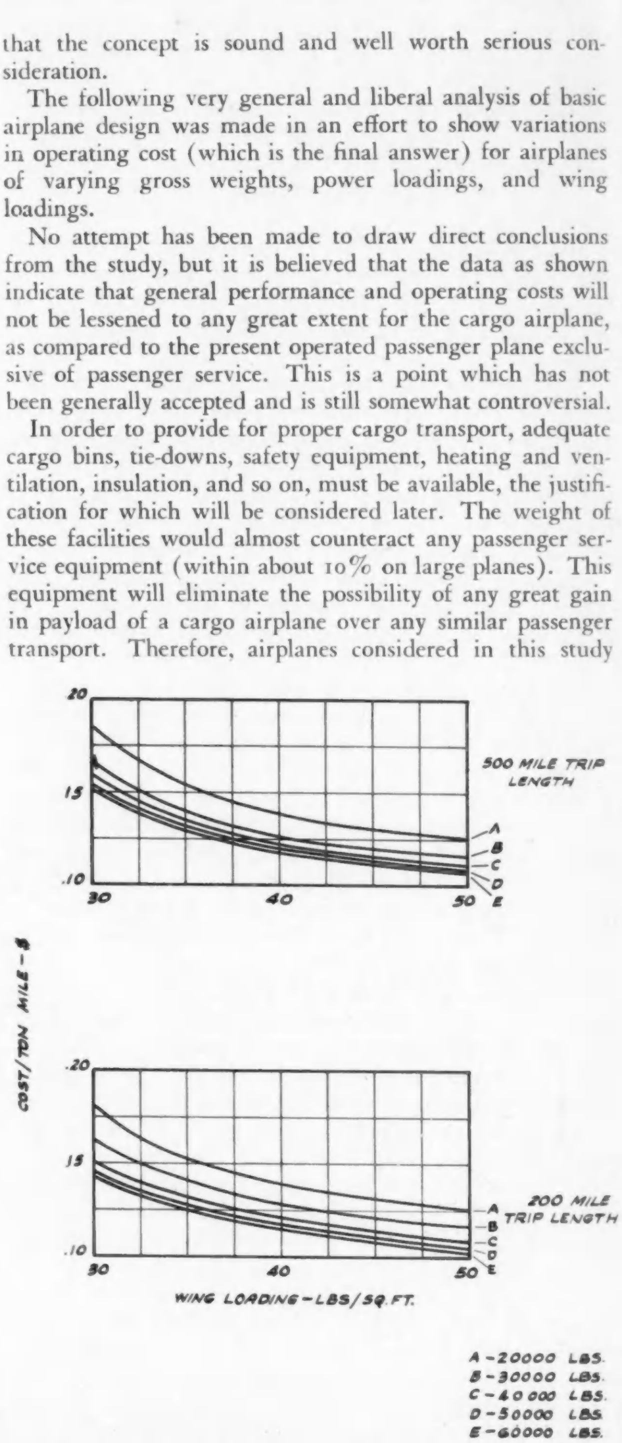
The small transport system need only adapt equipment as specified by Items (2) and (3) for its operations.

These basic thoughts are somewhat different from those usually expressed for cargo plane operation, but it is felt that the concept is sound and well worth serious consideration.

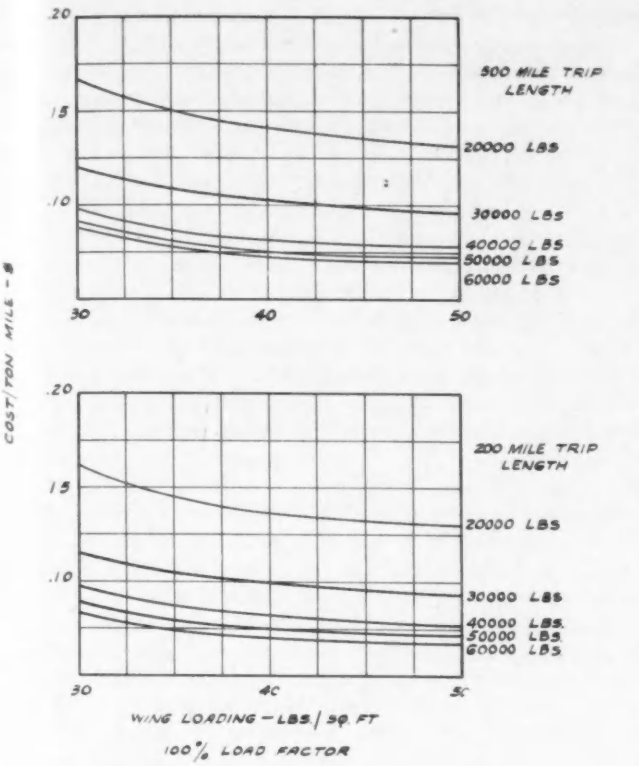
The following very general and liberal analysis of basic airplane design was made in an effort to show variations in operating cost (which is the final answer) for airplanes of varying gross weights, power loadings, and wing loadings.

No attempt has been made to draw direct conclusions from the study, but it is believed that the data as shown indicate that general performance and operating costs will not be lessened to any great extent for the cargo airplane, as compared to the present operated passenger plane exclusive of passenger service. This is a point which has not been generally accepted and is still somewhat controversial.

In order to provide for proper cargo transport, adequate cargo bins, tie-downs, safety equipment, heating and ventilation, insulation, and so on, must be available, the justification for which will be considered later. The weight of these facilities would almost counteract any passenger service equipment (within about 10% on large planes). This equipment will eliminate the possibility of any great gain in payload of a cargo airplane over any similar passenger transport. Therefore, airplanes considered in this study



■ Fig. 3 – Direct operating cost – $W/P = 8$ – 100% load factor



■ Fig. 4 – Direct operating cost – $W/P = 10$

have included the following equipment as being definitely needed and desirable: (1) supercharging equipment; (2) complete propeller and wing anti-icing equipment; (3) adequate heating and ventilating equipment; (4) 24-v electrical system; (5) full cabin lighting system; (6) cargo storage and tie-down equipment; (7) nose wheel; (8) flight engineers' station and equipment, on all planes with more than two engines. (If a flight engineer or flight mechanic is used on the smaller airplane, he may also act as a flight cargo handler, the duties for which are outlined later); (9) radio, including instrument landing units.

An analysis was made for conventional-type airplanes having the general characteristics shown in Table 1.

In order that no over-optimistic assumptions would be made with respect to payload, a thorough weight analysis was performed for the individual airplanes, the results of which are shown in the payload curves for 200 and 500-mile trip lengths (Figs. 1 and 2). The analysis was made for airplanes of from 20,000 to 60,000 lb gross weight, which appears to be the likely maximum range of cargo airplanes, based on present trends and studies. The curves show that the payload increases more rapidly for gross weight increases at the lower weights than at the higher values, though the slope becomes more nearly constant above 40,000 lb. This is true because additional engines and equipment (See Table 1) were used for higher weights, thus reducing the effective payload percentage. Also the same structural features were considered for all airplanes, whereas future engineering development may adjust this trend to increase the slope of the payload move.

The direct operating cost curves (Figs. 3, 4 and 5) indicate the already understood relation that increase in power gives increased cost, particularly when the payload percentage is decreased by the increased weight of the engines and additional fuel required to operate. It is interesting to note that the cost for the various gross weights at the lower power loadings cling closer together than for the higher loadings, due to several factors but primarily because of the lesser slope of the payload curve for the lower power loadings. It should also be noted that there is relatively little change in operating costs between the airplanes of various weights for the trip lengths investigated.

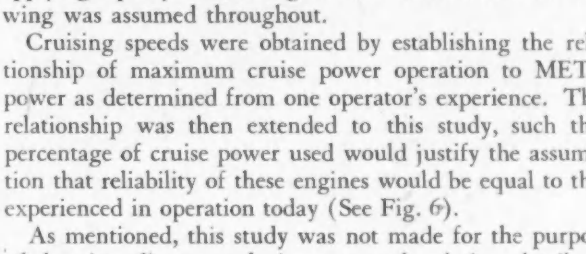
The study was expanded to include wing loadings of 30, 40 and 50 lb per sq ft, and power loadings of 8, 10, and 12 lb per hp. Power loadings were based on METO power (maximum except take-off power) conditions. When the METO power per engine exceeded 2000 hp, one more engine was used. This gave several three-engined airplanes as shown in Table 1. Although it is not the scope of this paper to discuss the relative merits of two-engine versus three-engine versus four-engine airplanes, it is believed that serious consideration should be given to the possibilities of three-engined equipment.

Though no rigid take-off and landing analysis was accomplished, sufficient study was made to indicate that the airplanes as shown should be capable of operation in and out of airports now served by airplanes of comparable sizes. The airplanes of high wing loading would require the use of an auxiliary high lift device. It must be remembered, however, that airports as used by an individual operator may dictate the power and wing loadings permissible, and choice would not be as wide as indicated here.

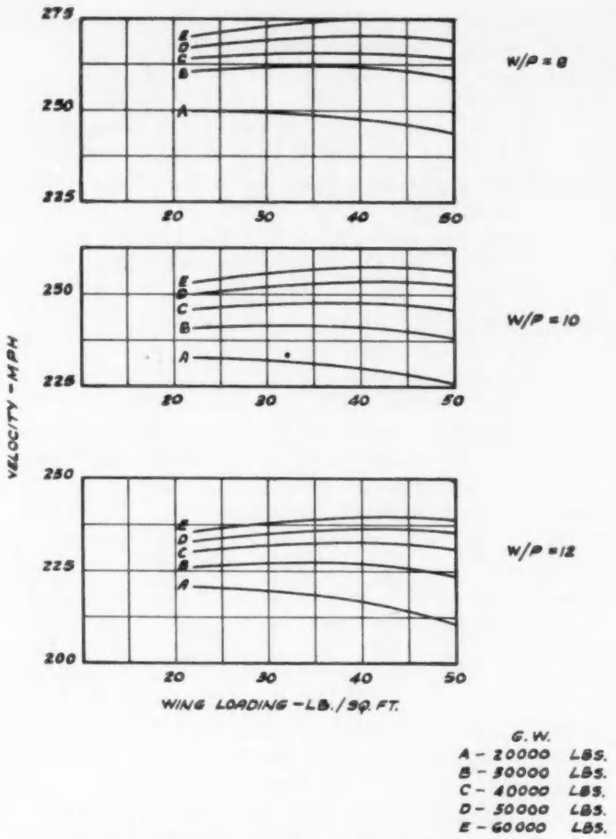
A complete study was made of drag characteristics and the best known qualities available were considered as applying equally to all designs. The NACA laminar-flow wing was assumed throughout.

Cruising speeds were obtained by establishing the relationship of maximum cruise power operation to METO power as determined from one operator's experience. This relationship was then extended to this study, such that percentage of cruise power used would justify the assumption that reliability of these engines would be equal to that experienced in operation today (See Fig. 6).

As mentioned, this study was not made for the purpose of drawing direct conclusions as to the design details of the most satisfactory cargo airplane, but is primarily for the purpose of stimulating thought on the general size and performance that might be required for cargo airplanes in



■ Fig. 5 – Direct operating cost – $W/P = 12$



■ Fig. 6 – Cruise velocity trend – 10,000 ft altitude

the immediate future, and to show very definitely that, discounting the present passenger service cost, direct operating costs of a cargo and/or passenger plane would be approximately the same.

B. Handling of Cargo Within the Airplane

The location of the cargo compartment is so involved with the design of the airplane that no set rule can be stated. Generally speaking, the centralization of roomy, accessible compartments near the center of gravity as practicable is recommended. Heretofore cargo space has usually been that space not easily usable for passengers, fuel, crew, radio or controls. The opinion is frequently expressed that cargo should be stored in wing compartments which would be accessible only on the ground. The writers wish to point out that outer wing space, where it can be utilized, is better suited to fuel storage or to the housing of permanent equipment. The center section portion of the wing of large airplanes can probably be used to advantage for major storage since it can connect directly with the fuselage space.

Neither do the authors advocate the use of the nose of the fuselage or the aft section for cargo storage unless they communicate directly with the main cargo space, and are used for special purposes (for example, lightweight cargo such as flowers traveling long distances, requiring a minimum of handling).

Doors and hatches merit major consideration in any airplane design, but in a cargo ship they are probably the most important factor in the control of ground time. Size and shape should permit the easy entrance and exits of packages and mail sacks. This requires the doors or hatches to be located as low as possible with respect to the ground. Approximately square or circular openings are considered most efficient. The present rear cargo door of the DC-3 will take 80 per cent of the present cargo.

Doors and hatches should communicate directly to areas within the airplane where sufficient floor space and head room exist to facilitate working. External doors and hatches should not open directly into a cargo pit, but rather into a passage way or working space. In large airplanes, cargo loading can undoubtedly be speeded up by providing cargo doors so that definite separate "flows" of incoming and outgoing material can be established. Long hinged doors, swinging down from the bottom of the fuselage have often been suggested as a means of providing easy entrance and a ramp. Such an arrangement is believed heavy from a structural standpoint and it probably prevents good use of the space above the door for storage or passageway during flight.

Floors, racks and bins have not usually been designed with any real thought for durability, utility and efficiency. Floors must be tight and have an ability to withstand the wear and tear to which they will be subjected. Tightness is important to prevent liquids, small bolts, wire, dirt, etc., from entering compartments (or control housings) below. Tightness is also helpful in cleaning and ease of cleaning is important. Probably a lightweight, tough, easily renewable covering for the floor is desirable, although much difficulty has been incurred in trying to obtain suitable companionway covering for the DC-3. A suitable covering material must be impervious to oils, alcohols, water and other common liquids. It should have a high friction coefficient, wet or dry, and must be able to withstand considerable scuffing and abuse.

Major bulkheads, if used in any given cargo plane design, will be fixed and generally unchangeable, because they will be utilized to carry stresses imposed by both flight loads and cargo loads. Therefore, any such major bulkheads should be designed to provide good passageway, headroom, and bin spacing. They should provide for the attachment and support of racks and bins. The design of the cargo airplane very definitely requires a full and adequate separation of the passenger section, if provided, from the cargo section. This separation must be flexible enough to permit the easy and quick removal of all the passenger equipment for adaptation of the airplane as a complete cargo unit.

Racks and bins should be readily adjustable to meet the needs of the particular operator. In many instances it will be desirable to vary the size of bins for different trips or flights over the same route. For example, the bin arrangement (and cargo distribution) within an airplane being routed out of New York westward into Chicago (with intermediate stops) might need be considerably different than for another airplane routed eastward out of Denver and into Chicago. The past practice of providing the biggest possible open cargo space in a given portion of the airplane structure involves a minimum of weight, but it does not promote efficiency in handling and segregation. Compartmentalization can be carried too far, but a moderate and sensible utilization of bins is surely advantageous.

The need for good temperature control, ventilation, lighting, and supercharging must not be overlooked by the designer. As stated at the beginning of this paper, all of these items must be taken into account in the transport of cargo by air. The need for temperature control is already recognized because flowers, certain foodstuffs,

birds, and so on, must be protected against freezing. Other items need to be kept cool for preservation. This calls for adequate heating and possibly cooling equipment, the control of which must be automatic. Passengers and crew will complain if they are subjected to an uncomfortable temperature, but cargo is not so helpful. Good temperature control requires the use of insulation and good ventilation. Airlines now have frequent difficulty with spoilage which results when perishable cargo is stored close to the skin of the fuselage and then covered with other cargo. Even though the air temperature of the compartment is kept above 32 F, the cargo may freeze when the airplane is flying at low temperatures. Likewise, cargo may become overheated when the airplane is standing in bright sunlight on a hot day. It should be noted that compartmentalization assists in avoiding this condition. Good ventilation is also necessary to eliminate odors, vapors, dampness, and so on.

Supercharging at least a portion of the cargo space will be essential, particularly those spaces where birds or other living creatures may be carried, or where relatively fragile airtight containers will be stored, which are not capable of withstanding considerable atmospheric pressure differences. This statement is made in spite of the fact that many designers have believed that supercharging could be eliminated entirely in the cargo airplane. They should not forget that mountains must be crossed, storms have to be avoided, and often high-temperature conditions and rough air will force the cargo airplane to fly at approximately the same elevations utilized by the passenger airplane.

Safety considerations for the cargo airplane must be equal to that provided for passenger transportation. At this time we are primarily considering the safety factors within the airplane itself and not those which result from operation through difficult weather conditions or over hazardous terrain. Of these safety considerations, fire prevention, detection, and control, stand out as presenting the most serious problems. Means must be provided so that any fire which can start during flight can be dealt with while the flight is in progress. The fire hazard will be decreased definitely by making all compartments readily accessible to the personnel and by the provision of adequate fire detection and extinguishing means. This is another reason for recommending against the use of remote cargo spaces such as would be provided in the wing structure or in the extreme ends of the fuselage.

Present air cargo is largely limited to those materials not readily combustible, but experience has shown that the fire hazard cannot be eliminated entirely by such selection. Therefore, one of the designer's chief safety problems should be to provide adequate protection which will permit carriage of all cargo now acceptable to railway express.

The shifting of cargo, during flight through turbulent air, or incident to a hard landing can create a very serious hazard. Here again compartmentalization and centralization are of definite assistance. In addition, means must be provided for the storage and securing of articles which are unduly heavy, and the shifting of which would likely cause any damage to the structure or injury to personnel. Adequate "tie-down" loops or rings must be available so that ropes or straps can be used to advantage in the larger spaces or where reinforced flooring has been provided for the carriage of heavy articles. If cargo airplanes are to make frequent stops, it is self-evident that they would be subjected to rough air conditions over a greater portion of their flight time, since the greatest amount of turbulence is usually encountered near the ground.

Liquids have not constituted a very large portion of air cargo to date, but consideration must be given to their carriage. Undoubtedly the amount will increase in the future. Since serums, lubricants, special chemicals, and so on, are often carried in liquid form, the designer must make certain that the spillage of these liquids will not constitute a hazard. Special packaging will undoubtedly be necessary, but even then it must be made sure that the spillage of any liquids will not cause damage to any great amount of the cargo, cause corrosion to other parts of the airplane, or create fumes or odors which will be injurious. Probably some means should be provided for the disposal during flight of packages which contain damaged liquid containers.

The possible fouling of controls, tubing, conduit, and so on, has always been a matter of serious consideration by air transport operators, and the problem will be even greater with cargo airplanes where heavy packages will be handled and where all cargo is likely to be associated with more or less dirt and dust. Therefore, protection of controls, tubing, electrical conduit, and so on, must be given added attention. They should be entirely enclosed so that material cannot be thrown into them either during loading or in flight. Short circuits caused by damage to electrical wiring constitute definite fire hazards and possible loss of use of radio and electrical equipment. Damaged fuel or oil lines can result in loss of power or control and injury to cargo, and a serious fire hazard can be created thereby.

The cargo plane designer is faced with the necessity of developing an airplane which will not be critical to moderate variations of the center of gravity. In the sorting of cargo and during rapid loading and unloading, it will be impossible closely to control the center of gravity, due to the wide variations in type of material to be carried from day to day. Special provisions will have to be made for the extremely heavy or extremely large parcels, but nevertheless, much ground time will be saved if the need for precise weight distribution can be avoided. This appears to be another valid reason why cargo space should generally be disposed symmetrically about the center of lift of the airplane. The present DC-3 airplane has a fairly good range of balance limits, but it is believed a greater spread should be planned for any new cargo plane design. In the case of the DC-3, fair flight characteristics are maintained for a variation of center of gravity from 11% to 28% of the mean aerodynamic chord.

Accessibility to cargo compartments during flight has been mentioned from the standpoint of safety, but it is also important because of the probable need for handling and sorting cargo during flight. Doubtless there are many who look upon the actual working of cargo during flight as something to be reckoned with far in the future, but it seems quite possible to the writers that the time for such procedure is not far distant. It may not be necessary to have a cargo handler on board at the originating point of a trip, but one may likely be required to work in and out of other major terminals where the exchange of cargo will be large and the time short.

For instance, with a westbound trip originating at New York City, sufficient time should be available prior to departure to load and segregate all of the material in an efficient and orderly manner. In making a stop at Cleveland, there should be a fairly large exchange of cargo; it may be advantageous to put a cargo handler on board to

sort and distribute packages enroute to Toledo. A rapid exchange of cargo could then be accomplished at that stop, because the cargo handler would have all "off" material immediately ready to unload. It would only be necessary to load the "on" material into a central compartment from which the cargo handler could make further distribution during the flight to Chicago. Here again he would have the "off" material ready for rapid unloading and the "on" material would be placed aboard the airplane with perhaps only partial distribution, so that time on the ground would be saved.

The possibility of working mail should also be considered, although no direct recommendations concerning facilities for such are being made in this paper. It is believed that, if the possibility is kept in mind, it will work well into the design features which are being recommended.

The avoidance of cramped working quarters for ground or flight crews is essential for efficient handling of material. Present airplanes do not provide this feature. In many cases the crews have to work directly in the cargo compartments, and to be in a stooped or squat position to utilize the space provided. When packages can be worked in only one direction at a time, the rate of loading or unloading is limited to the speed of the slowest individual in a cargo passing line composed of two to four or five individuals. This is reminiscent of the old bucket brigade of fire fighting and is just about as efficient.

The arrangement of compartments should be such that any package can be reached promptly. This arrangement calls for an efficient marking of compartments and a set of records from which the cargo handler can quickly ascertain the location of any package. This involves paper work which should be simplified as much as possible, but which must be adequate so that load and balance can be controlled.

During the course of the preparation of this paper, the authors have come to the conclusion that standard packaging would be an immense assistance for orderly sorting and handling. So far as it is known, it has not been proposed heretofore for air cargo. If a basic standard package size of, say 6 x 6 x 6 in. could be established, and then other standard packages designated which would be multiples of these dimensions, it should be much easier for the designer to set up efficient compartments. Further, the job of the cargo handler would be much simplified because he would know exactly how to load the material into his bins with the greatest amount of efficiency. If such a scheme meets with approval, it would seem reasonable that the airlines and the express companies should decide upon a number of sizes of standard packages which would be multiples of the basic dimensions selected. Specifications should be written to cover the type of materials to be used in such packages and to give other packing instructions. It would seem advisable to encourage the use of such standard packages by placing a premium upon the carriage of shipments which were not placed in a standard approved package. The authors do not consider themselves competent to go deeply into a standard packaging analysis, but felt that it should be suggested because, if adopted, it will have an important bearing upon the design of the cargo airplane.

C. Equipment External of the Airplane

If a cargo airplane is to be unloaded and loaded rapidly during very short periods of time on the ground, it is essential that all ground-handling equipment and methods be extremely efficient. In the first place, much study and thought must be given to a means for sorting and handling cargo as it comes into the terminal from all of the various sources. This arrangement will require a special building or room which will be used in common by all of the carriers. Such spaces are now jointly used at a good many of the terminals to good advantage, but it should be pointed out that the necessity for transferring the cargo from an incoming airplane to the clearing room, and then to return that cargo to another airplane scheduled for immediate departure, involves a serious loss of time. While such a clearing house or storage space will be necessary and will be used to advantage when any appreciable layover will be involved, it is apparent that provisions must also be made for quick transfer of cargo directly from one airplane to another.

Trucks, racks, carts, and so on, which are used for transportation and loading of cargo must be designed to be universally applicable to the different airplanes. At the present time every airline has its own design of equipment, some of which is very good and some is very poor. Enough experience has now been gained so that the better features can be picked out and consolidated into more efficient units. The design and building of adequate cargo-handling equipment provide a good field for the manufacturer who is willing to concentrate on the problem.

Various types of parcel carriers and lifts have been proposed so that packages can be hoisted easily to cargo doors which are at a considerable elevation off the ground. As previously shown, it is desirable that future airplanes be designed so that the cargo doors will not be high off the ground, but there may still be instances where hoists will be advantageous. Such hoists and carriers should eliminate most of the "bucket-brigade" type of transferring parcels from one man to another, which is now common practice and which means that every parcel is handled several times in the transfer. All trucks, racks, carriers, and so on, need to be designed so that they will not be overturned or unduly shifted by blasts of air produced by the propeller or by windstorms.

Wind breaks and canopies must be provided to protect the cargo handlers and to avoid damage to the cargo from weather. At the present time it is not unusual for identification stickers and waybills to be lost due to the wind and frequent handling.

Proper illumination of trucks, carts, and so on, is essential to facilitate easy and rapid identification and routing. A well-lighted and protected space should be provided where the cargo handler can check off the packages as they come out of and go into an airplane. Such space should also be properly illuminated and heated to reduce errors to a minimum.

Since it is so essential to keep ground time to an absolute minimum, much pre-planning of the loading of a given airplane should be accomplished. This may call for sorting bins on the trucks and improved methods of identifying and routing of parcels. Possibly the trucks should be provided with cargo compartments similar to those within the airplane so that cargo can actually be segregated and distributed in a manner similar to that it will have during flight. Such pre-planning and prearrangement would undoubtedly reduce the possibility of last-minute shifting and changing, and should contribute towards better balance and weight control.

Limiting Factors of OVERHAUL PERIODS for AIRCRAFT ENGINES

by MARVIN WHITLOCK
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FOR the purposes of this discussion, a hypothetical engine was adopted which is intended to possess operating characteristics representative of all of the types and models of engines currently being operated by the domestic airlines.

The desirable or optimum overhaul period is described as being that interval during which the engine can be operated with maximum economy and without sacrificing the reliability of any other overhaul period.

A brief description is given of the various groups of engine parts and a typical overhaul inspection of the pertinent parts is described. It is concluded that the limiting factor for engine overhaul periods is the accumulation of the centrifuged deposit in the crankpin which limits the overhaul period to approximately 1000 hr. It is further concluded that this condition can be alleviated by the provision of effective filtering facilities in the lubrication system and by improved scavenging of the lead products of combustion.

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THE purpose of this paper is to discuss the factors which regulate the frequency of aircraft-engine overhauls in an effort to establish the limiting factor or factors applicable to the overhaul periods of engines being operated by the domestic airlines at the present time. It is not intended that this discussion compare engines of different manufacture or models, but that it describe a hypothetical engine having service characteristics representative of the average currently operated engine. Therefore, the service difficulties and service life figures quoted herein are not necessarily common to all makes and models of engines and, in some cases, will impose a penalty on one manufacturer. However, in other cases, the quoted figures will be complimentary, and it is therefore expected that this discussion will depict the average conditions as they exist today.

While the discussion concerns specifically the commercially operated aircraft engines, it will also apply in part to engines of comparable types operated in military service.

The relation between the engine manufacturer and the operator is that they are both equally and primarily interested in reliability. However, the engine manufacturer, as his experience and judgment permit, is interested in increasing the power rating of the engine by making a minimum of changes to the engine since horsepower-per pound weight is the basic commodity that he has to sell. At the same time, the operator, as his experience and judgment permit, is interested in increasing the length of time between overhauls since the prime justification for the air carriers' existence is the speedy transportation of passengers, mail, and express at a profit.

As an illustration of the re-rating of an engine, Fig. 1 shows the increases in output of a 9-cyl single-row engine. The engine was originally released in 1928 as a 525-hp engine and the rating has been increased from 525 hp to 1200 hp in 11 yr without altering the basic dimensions of the engine.[1] At the same time, the weight in pounds per horsepower has consistently decreased from 2 lb per hp in 1928 to 1.07 lb per hp in 1939[1]. On the surface, this appears to be progress; however, it is not progress unless the service life of the parts are proportionally increased and this has not been true in every case. Actually, the only justification for a manufacturer increasing the power of an engine is that the new engine will afford equal or improved reliability and equal or lower maintenance and operating cost.

A typical history of overhaul extensions by an operator is shown in Fig. 2. It is general practice to start the operation of a new model engine at a conservative overhaul period of 300 to 400 hr. This practice is not to be construed as a "break-in" or "run-in" period for a newly manufactured engine but is instead a service test period for a new model engine. The subsequent rate at which the overhaul periods are increased depends, of course, on the amount of reliable and satisfactory service obtained, and on the condition of the parts at time of overhaul.

An extension in engine overhaul period requires a revision to the operators' Certificate of Competency which is issued by the Civil Aeronautics Authority. Prior to an application for an extension, a report is compiled furnishing detail data regarding any failures that may have been experienced, rates of wear and general condition of the parts at the time of overhaul. The report is filed with the

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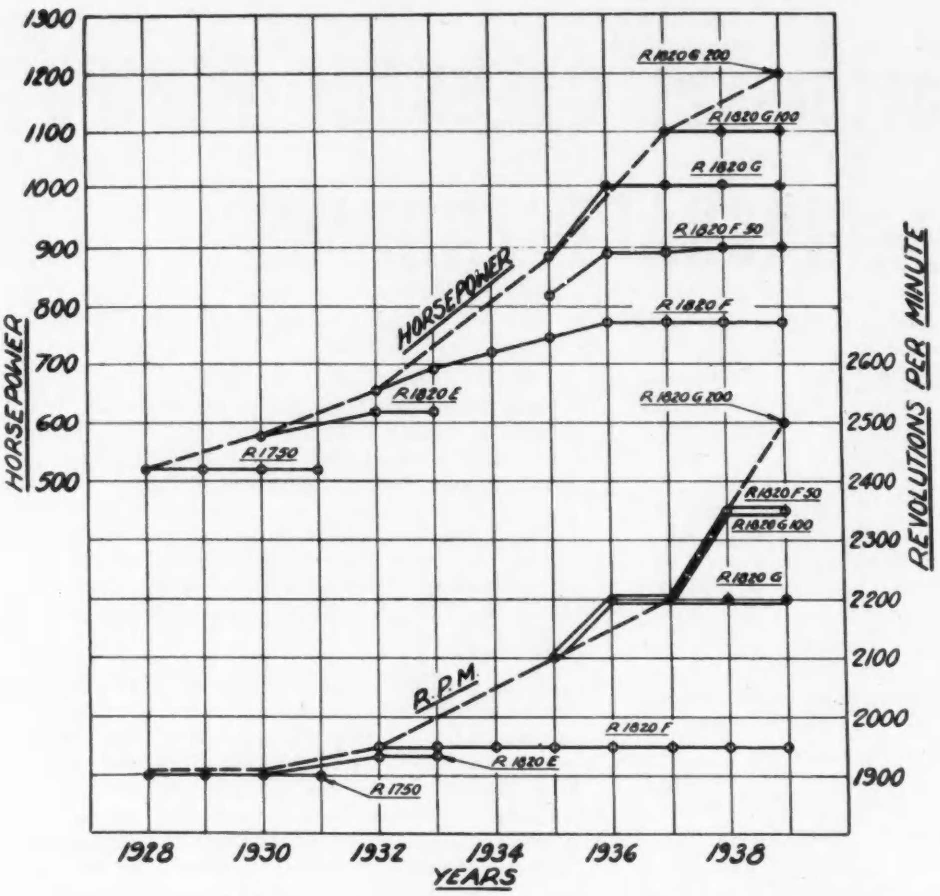
[1] See SAE Transactions, January, 1940, pp. 18-24: "Design Problems in the Quantity Production of Aircraft Engines," by Henry C. Hill.



■ Fig. 1 – Development of Wright Cyclone engine[1]

hydraulic pump, oil cooler air intake, oil temperature regulator, tachometer magneto exhaust collector ring, booster coil, the necessary plumbing and wiring, and suitable cowling and baffling to cool the prime mover with a minimum of aerodynamic loss and to prevent spread of engine-originated fire to the aircraft structure. This assembly is removable from the aircraft as a unit and it is the overhaul of this assembly that is referred to by the term "engine overhaul."

Due to the physical limitations of this paper, it is necessary that we limit this discussion to the prime mover or bare engine, and it can be said that the previously enumerated accessories will not be limiting factors to the overhaul period, providing they can be removed without disturbing the engine in its mount.

We therefore adopt for discussion the hypothetical engine previously mentioned. This engine may be described as being a 9-cyl single-row or a 14-cyl double-row, radial, aircooled, reduction-geared engine, of approximately 1825 cu in. displacement, having a single-speed supercharger and rated at 1000 to 1200 hp for sea-level take-off. The engine will be as manufactured either by Pratt & Whitney Aircraft or Wright Aeronautical Corp.

There are some 900 such engines in current commercial use in this country, and their periods of operation are logged in hours and minutes dating from terminal to terminal. After 675 hr, which is the approximate current average of all types and models of engines, this engine is removed to be overhauled by the operator in a manner prescribed by the engine manufacturer.

The question of "What is the limiting factor for time between overhauls of aircraft engines?" is indeed a moot one and, if this question were asked of the average engine overhaul foreman or manufacturers' service man, the extemporaneous answer would probably be "courage" or "confidence." In a sense, this is a logical answer as the courage or confidence is inspired by factual evidence or visual witnessing over a period of time of the satisfactory performance of an engine and of the condition of the parts after a given length of service. Obviously this extemporaneous answer cannot be accepted as the true explanation, and we will endeavor to arrive at the proper conclusion in the following discussion.

No doubt the question has arisen in your minds as to what is the optimum engine overhaul period. What do the operators expect of an engine? The answer to these

Civil Aeronautics Authority who examines the data and either approves or rejects the request as the evidence warrants.

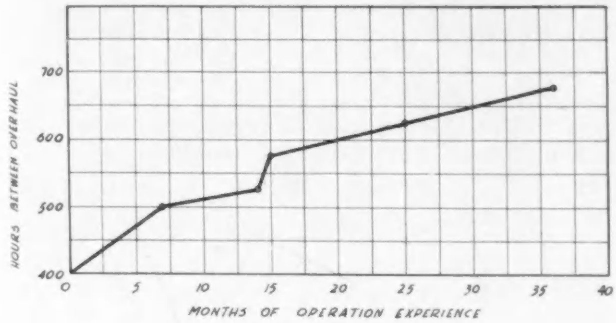
Fig. 3 is presented to assist in evaluating the importance of the subject. This figure illustrates the manner in which the maintenance expenses of a typical airline are divided. This figure represents the labor, material, and supervision necessary to maintain all equipment required for air-carrier operation and does not include the purchase of new equipment, depreciation, fuel and oil or any expenses involved in operating the equipment. In monetary figures, there was an estimated $16,000,000 spent by the domestic operators in 1941 for airline maintenance. Of this amount, there was some $5,000,000 devoted to engine maintenance and overhaul of which some $2,750,000 was spent for engine overhaul. It is obvious from these figures that the overhaul periods of aircraft engines merit consideration not only because of the possibility of opportune financial benefits but also, what is more important at the present time, the need for producing and conserving horsepower. Increasing the service life of engine parts is equivalent to increasing the industry's production.

From the interest of an airline maintenance department, the engine is referred to more properly as a powerplant assembly and consists of the prime mover or engine proper together with the accessories essential to the operation of that prime mover and to the aircraft. Such an assembly will normally consist of an engine mount, propeller, propeller governor, starter, generator, carburetor, air intake, fuel pump, two magnetos, ignition harness, vacuum pump,

questions is that the optimum or desired overhaul period for a given type service is that period during which the engine can be operated at a minimum maintenance cost per hour and with maximum reliability; in other words, with reliability at least equal to that of any other overhaul period. The industry desires and strives for 100% reliability regardless of the number of engines on the aircraft and regardless of the price that is attendant. Although 100% reliability is never realized, it is never knowingly sacrificed for lengthened overhaul periods.

This optimum or desired overhaul period may not necessarily be a long period for a given engine, due to economic reasons. Assuming that the reliability remains constant, the economic savings attendant to extensions of overhaul periods are derived principally from two sources: first, the direct saving in labor; and second, the savings in replacements parts costs.

The saving in labor is by far the more tangible of the two sources, although it is not necessarily the more valuable of the two. Available data indicate that the average



■ Fig. 2 – Typical rate of extensions in engine overhaul periods[2] (Pratt & Whitney S1CG and S1C3G engines – United Air Lines Transport Corp.)

engine overhaul labor and material cost is approximately \$846 per overhaul for a 675-hr overhaul period of which some \$170 is allocated to direct labor. On this basis, a 100-hr overhaul period increase would result in a saving of \$25 per engine overhaul for labor alone.

The economics of parts replacement costs as related to extended overhaul periods is much more complicated because the savings are dependent on the effect of the length of overhaul periods on the durability of the parts. The service life of some parts will not be affected by extensions of overhaul periods; others may have shorter service lives and still others may have longer lives. For those parts which are not affected, the most economical overhaul period is that period which is equally divisible into the service life figure of the majority of the more expensive parts since the high initial costs of the parts makes it particularly desirable not to sacrifice any more "life remaining" than is necessary. For those parts whose service lives are affected, the most economical overhaul period can be determined only by evaluating the gains or losses due to the effects of the overhaul period extensions.

This analysis is somewhat theoretical and is exceedingly hard to apply to a given engine; however, it is indicated that the limiting factor to the optimum or desired overhaul period may be an *economic limit*. However, it is very doubtful that this "economically limited" overhaul period would be reached before a "mechanically limited" period would be encountered and we therefore direct our attention to the mechanical factors which might cause the overhaul period of current engines to be limited.

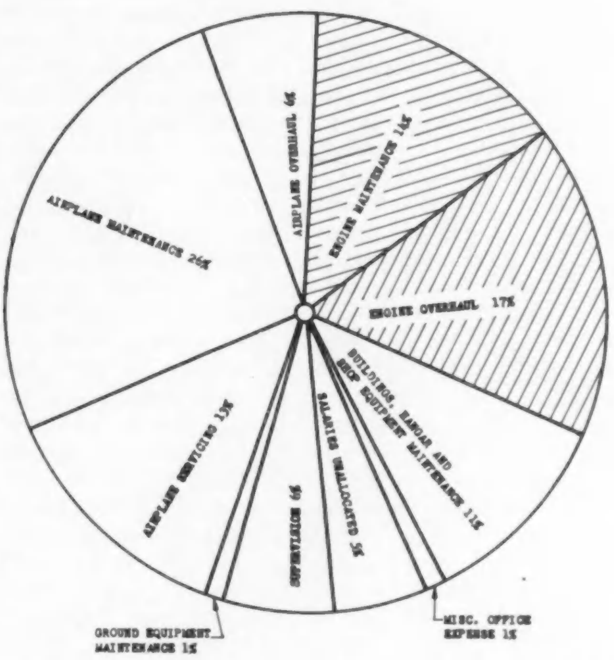
To assist in this analysis, the writer feels it desirable to discuss briefly the conditions that would be looked for at time of overhaul inspection of our "hypothetical" engine and the expected findings based, of course, on the consensus of current experience. This procedure can be facilitated by grouping the major portions of the engine and discussing them as such. It is understood, of course, that this does not describe a normal overhaul inspection of an engine but only the portions of the inspection pertinent to this discussion.

■ Engine Overhaul - General

At the termination of the number of hours of operation established as the overhaul period, the powerplant is removed from the aircraft as an assembly and a newly overhauled assembly is installed. The newly overhauled engine already will have completed at least 6 hr of test-stand operation and, after installation, will have miscellaneous control adjustments as indicated to be necessary by the ground check. It will have the propeller-engine combination dynamically balanced and then will be test flown prior to its being put into service.

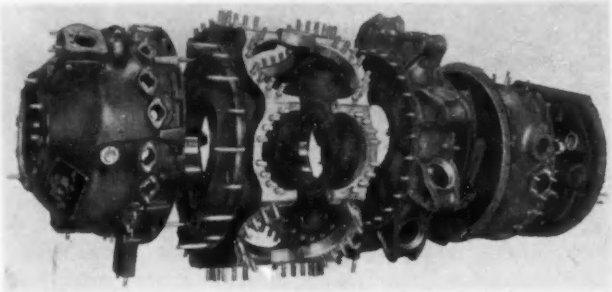
The assembly removed from the aircraft is disassembled into subassemblies which are routed to their respective departments for cleaning, inspection, repair, and final inspection. For example, the cowling is removed, washed, and sent to the cowling repair shop for inspection and repair. The same is true for the engine mount, propeller, accessories, fuel and oil lines, and other parts of the assembly which have been enumerated previously.

The engine, as referred to in this discussion, is disassembled and cleaned according to procedures approved by the engine manufacturer. This procedure involves the washing of certain parts in cleaning solvents to remove oil and sludge accumulations and, in other cases, involves



■ Fig. 3 – Typical allocation of airline maintenance expenses

[2] Letter of Dec. 8, 1941, from R. D. Kelly, United Air Lines Transport Corp., to the author.



■ Fig. 4 – Reduction-gear housing, crankcase, blower section and rear cover (Wright Cyclone[3])

sand blasting of the parts to permit reliable inspection. The parts are then forwarded to the inspection department which is equipped with magnaflux equipment and the necessary jigs and indicators properly to inspect and record the condition of the various parts in accordance with procedures prescribed by the manufacturers.

■ Reduction-Gear, Crankcase and Blower

The reduction-gear housing (frequently called the nose section), crankcase front section, crankcase rear section, supercharger front section, supercharger rear section and rear cover plate are shown in Fig. 4. The parts shown are typical for a 9-cyl, single-row engine. The double-row engines have an additional or centre section to the crankcase. These parts collectively serve as the frame or housing for the engine.

The detail design and the material used vary with the model engine, but it is considered typical for the reduction-gear housing to be made of forged aluminum alloy, the crankcase of steel or forged aluminum alloy, and the supercharger sections and rear cover of cast-aluminum alloy or magnesium.

The stationary reduction gear and the thrust bearing are both located in the forward end of the reduction-gear housing. This arrangement requires that the reduction-gear housing transmit the torque reaction and the thrust loads to the crankcase proper. The crankcase supports the cylinders and the crankshaft main bearings as well as to transmit the thrust and torque reaction loads from the nose section. The supercharger section attaches to the crankcase and is provided with motor mount attachments which support the engine in the airplane. This section houses the blower and is equipped with individual cylinder intake pipes radiating from the annulus formed at the outer portion of the section. The rear blower section provides a vaned diffuser plate at its forward face, houses the blower and accessory drive trains, and provides a mounting for the downdraft carburetor. The rear cover provides a mounting for the accessories, bushings for the accessory drives, and drilled oil passages for lubrication of the accessory drives and as part of the lubrication system.

The service life of these parts is limited only by their endurance limit. The forged aluminum-alloy reduction-gear housing, steel crankcases, and cast-aluminum-alloy blower sections will operate satisfactorily for 8000 hr and suffer only failures due to fatigue which is indicated by cracks usually around studs, holes, or fillets. The magnesium rear covers have a life of approximately 3000 hr and may finally fail due to fatigue. Quite a few heavy accessories are supported by this rear cover.

The portions of these parts that are subjected to wear are protected by replaceable retainers or bushings, the majority of which are in the rear cover for the accessory drives. These bushings are pressure-lubricated and relatively lightly loaded. The rates of wear are therefore low and the service lives are upward of 6000 hr.

Some corrosion is experienced on the thrust bearing retainers, and some chafing is found occasionally between mating surfaces, but neither of these conditions is serious.

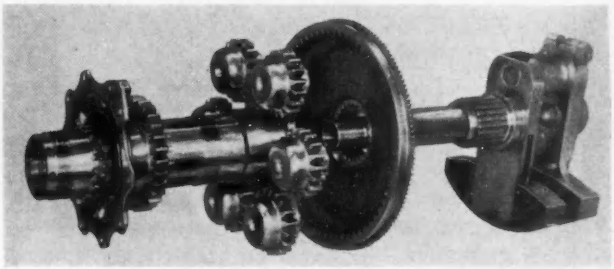
The accumulation of sludge, carbon flakes, and leak in the oil passages and on the walls is not serious during a 700-hr period, but is progressive and might become serious during extended periods.

We therefore conclude that, in this group, the rear cover has the lowest service life, and it is believed that this would also be its maximum overhaul period. The accumulation of lubrication deposits would be an allied limitation at that time or possibly before.

■ Propeller Shaft and Reduction Gear

The details of the reduction gear vary with different model engines but, in general, the assembly consists of a stationary or sun gear splined or bolted onto the nose section, a number of planetary pinion gears attached to the legs of a spider which is integral with the propeller shaft. The pinion gears mesh with the stationary gears and with a bell gear which is splined to the crankshaft. See Fig. 5. The pinions are fitted with steel-backed leak-bronze bearings; the shaft proper is tubular and is splined in accordance with SAE standards to accommodate the propeller.

The pinion-gear bearings are pressure-lubricated by means of drilled passages in the legs of the spider, and the remainder of the assembly is splash-lubricated.



■ Fig. 5 – Crankshaft, propeller shaft, and reduction-gear assembly[3]

The shaft is inspected for cracks, lubrication-passage cleanliness, and condition of the splines. The gears are checked for pitting, wear, and cracks.

The service life of the shaft proper is limited only by its endurance limit which is upwards of 9000 hr on current engines. It is very rare that there is any wear on the splines. The oil passages are generally found to be clean.

The only portions of this assembly that are subject to wear are the pinion bushings and the gear teeth. The tooth wear on the bell gear limits its life to approximately 5000 hr. Considerable pitting has been experienced on the stationary and the pinion gear teeth. This pitting is credited to lack of uniform hardness gradient resulting in subsurface fatigue failure and has been prevalent above 1500 hr of service.

[3] Instruction Book for Wright Aeronautical Corp. Cyclone engines.

We therefore conclude that the limiting factor for this group is the pitting of the gear teeth which limits the period to approximately 1500 hr. The assembly is also vulnerable to lubrication-system deposits and this would be a questionable factor if operated for 1500 hr between overhauls.

Crankshaft

The crankshaft design varies considerably with different models and types of engines but, in general, it consists of a single- or double-throw, counterbalanced, steel-forging crankshaft supported by roller bearings and provided with a hardened steel crankpin-journal. See Fig. 5. Torsional vibration damping is accomplished by the manner in which the counterweights are constructed or attached.

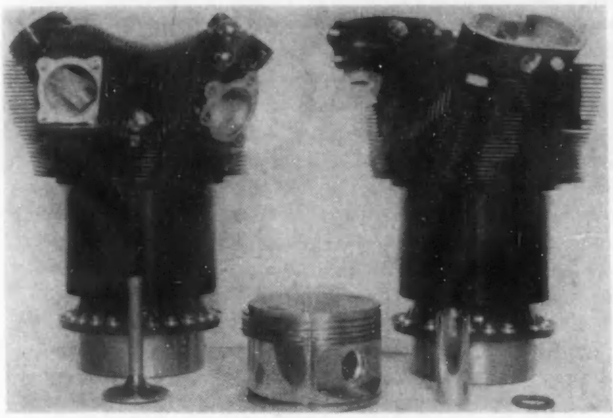


Fig. 6 – Wright Cyclone G-200 cylinders, piston, and exhaust valve[4]

It is general practice to use the crankshaft to convey lubricating oil from the rear of the engine to the crankcase section. A cavity is provided in the crankpin and forms part of the lubricating system in that it acts as a passage and as a centrifuge. Outlet tubes extend from approximately the center of this cavity to flats on the crankpin bearing surface in such a manner that the tubes transmit supposedly clean oil from above the surface of the "cake" formed in the crankpin cavity due to the centrifuging action to the crankpin surface. This oil lubricates the crankpin journal and master-rod bearing, lubricates the knuckle pins through drilled passages, and lubricates the power section of the engine by means of controlled leakage at the ends of the master-rod bearing. Usually there is also a jet located near the end of the crankpin which furnishes oil for the power section of the engine.

At time of inspection, the crankshaft is magnafluxed to check for cracks particularly around splines, oil holes, and fillets. It is very rare that a crack will be found as fatigue failures of crankshafts are almost unknown.

The wear is checked on the crankpin and at the oil-seal rings in the tailshaft couplings. Wear at both of these places is particularly important as it affects the lubrication of not only these parts but of a number of other parts.

The wear on the crankpin will average less than 0.001 in. on the diameter for a 700-hr period and this rate of wear limits the life of the part to approximately 7000 hr. Wear on the oil-seal ring surfaces requires that the surface be replated approximately every 2000 hr. Very little wear is experienced on the splines.

The assembly is inspected for cleanliness of the oil passage and a cake of silicious, carbonaceous matter will be found in the crankpin cavity. This cake will usually fill from one-third to one-half the volume of the cavity and will weigh approximately 10 to 13 oz. As the deposit increases, the crankshaft is thrown out of balance, the centrifuging efficiency decreases and, as the surface of the cake approaches the top of the oil tubes, the oil provided to the bearing no longer has the advantage of what centrifuging action the crankpin does offer.

This deposit is an outstanding limitation to the time between overhauls with the present oils and lubrication system, and it is believed that the accumulation will be excessive within 1000 hr. of operation.

Connecting-Rod Assembly

The connecting-rod assembly consists of one master rod and the required number of link rods per bank of cylinders. The rods are of I-section construction, are made from special alloy steel forgings, and are machined and polished on all surfaces to a very high standard. The link rods are attached to the master rod by means of knuckle-pin joints and to the pistons by means of piston pins. A bushing, usually of bronze material, is provided in each end of the link rod to provide a bearing for the attachment to the piston pins and to the knuckle pins. The master rod may be of a one-piece or a two-piece construction using a steel-backed lead-silver or lead-copper bearing with or without a lead flash.

As explained in the discussion on crankshafts, lubrication oil is provided under pressure to the master-rod bearing and to the knuckle pins through drilled passages in the master rod. The full-floating piston pins are lubricated by the mist of oil thrown from the oil jet and from the end of the master-rod bearings.

These parts are magnafluxed and inspected for cracks, straightness, wear on the bushings, the bearing, and for cleanliness of the oil passages.

The wear on the link rod bushings is very low, and these parts have a service life of approximately 4000 hr.

The service life of the master rod and link rods is limited only by their endurance limit and, due to the design and workmanship on these parts, this limit is upward of 8000 hr.

The oil passages are not extensive and are generally found to be clean.

The limiting factor of the overhaul period on this assembly is believed to be the master-rod bearing wear which is particularly important since the amount of oil thrown into the power section with the subsequent heat rejection influence is very sensitive to bearing clearance. A number of failures have recently been experienced on the lead-flashed, copper-lead type bearings, and the time between overhauls of this part is limited to approximately 1500 hr.

Cylinders, Valves, and Valve Springs

The cylinders in current use are of steel and aluminum construction. The cylinder barrels are machined from steel forgings and have integral cooling fins. See Fig. 6. The head, including the fins and rocker boxes, is cast of aluminum alloy, and in one piece. The head is screwed and shrunk onto the barrel. Each cylinder has one inlet

[4] See "Engineering Aspects of the Wright Single-Row Cyclone G-200 Series Engine," by H. C. Hill.

and one exhaust valve seating on a bronze seat in the case of the inlet valve and on a hardened steel seat in the case of the exhaust valve. The seats are shrunk into the head casting. The intake is a tulip design valve and the exhaust is a hollow head and stem design valve using sodium as a coolant and incorporating a stellite face. The cylinder walls are splash-lubricated and the valve is theoretically lubricated by oil from the rocker boxes. Each cylinder has two spark-plug bosses, one front and one rear.

The cylinder is magnafluxed and checked for cracks in the barrel. The head is checked visually for cracks after sandblasting. It is very rare that any cracks will be found in the cylinder wall; however, it is common experience to find incipient cracks around the seat inserts, radiating from the spark-plug bosses, and in or near fins. It is considered practical to re-barrel a cylinder head but not to put a new head on an old barrel; therefore, cracks in the head limit the service life of the cylinders on many engines to approximately 3000 hr.

Cylinder-barrel wear varies with the type of engine due to the piston, ring and cylinder design and due to other factors such as fuel, amount of power used, and type oil but, in general, the wear is such that it will limit the life of the cylinder to approximately 3000 hr.

Valves and valve guides normally have a service life of approximately 3000 hr; however, considerable burning is being experienced on the exhaust valves, valve seat, and cylinder end of the valve guide, and of the valves sticking in the guides. As will be shown in the discussion, these deficiencies are not necessarily a function of time since overhaul.

It is believed that wear on the cylinder barrels is the most serious limitation to extensions of the overhaul period of this assembly and that this limit would not be in excess of 3000 hr and this length of time would be subject to check as to the effect on the rate of wear if the cylinders were not rehoned or relapped at intervals.

■ Pistons and Rings

Pistons in current use are made of aluminum and usually have five rings above the piston pin and one below the pin. See Fig. 6. The ring below the pin is a "pumping" ring instead of a scraper ring and two drained oil scraper rings are provided above the piston pin. The three upper rings are wedge-shaped in cross section to eliminate sticking and are provided with a tapered face. This type piston is referred to as a "uniflow" piston and, as you have surmised from the description, is intended to keep the skirt area well lubricated. This design has accomplished its purpose with notable results.

The pistons are inspected for burning, cracking, wear, corrosion, carbon accumulation in the grooves, and for sludge accumulation on the underneath side of the piston.

Very little burning will be found, providing the engine has not been detonating and practically no cracking is experienced. The rate of wear on the piston is very low, and the amount of corrosion found on the skirts is not serious.

There is some carbon in the ring lands; however, with the wedge-type rings, it does not cause sticking. Although, since this build-up is a function of time, it is believed that the build-up would be excessive within 1500 hr of operation.

There is an abundant collection of carbon or sludge usually found on the under side of the piston. This accumulation does no direct harm, but it does provide a prolific source for carbon flakes which may cause clogging of passages, and operation for periods of longer than 1500 hr would be subject to extreme discretion from this standpoint.

The ring sticking so prevalent a few years ago has been superseded by a very vicious "pounding out" of the upper side of the top ring land. This limits the life of an otherwise good piston to approximately 2000 hr.

Ring wear limits the life of the rings to approximately 1500 hr, and the ring wear, together with the carbon and sludge accumulation, limits the overhaul period of this assembly to approximately 1500 hr.

■ Cam and Valve Actuating Mechanism

The operation of the valve is accomplished by means of a mechanism consisting of a cam and cam drive gear, tappets including rollers and guides, push rods, rocker arms, and valve springs.

It is general practice to make the cam as a single piece, double-track hardened cam ring which is bolted onto a light-weight hub provided with a bronze bearing and driven from the crankshaft by means of one or two sets of gear and pinions.

The valve tappets incorporate a hardened roller as a cam follower and are a close sliding fit in the guides which are attached radially around the nose section or part of the case.

The push rods are made of steel tubing with hardened steel balls pressed into each end.

The rocker arms are made from steel forgings and are mounted on a double-row roller bearing, the inner races of which are clamped in the rocker box by a through bolt. The valve end of the rocker arm incorporates a hardened roller, and the push-rod end is cupped to fit the push rod and provides means of adjustment for clearance.

The valve springs consist of either two or three concentric coil springs. One end of the springs seat on the shoulder of the valve guide and the outer end is attached to the valve stem by means of a shouldered washer and a tapered, split lock ring.

The cam drive gears and the cam bearing are pressure-lubricated, and the cam surface and tappet rollers are splash-lubricated. Oil is provided to the tappets which are designed so as to act as a reciprocating pump to pump oil through the push rod into the rocker arm where it lubricates the rocker-arm bearing and the leakage lubricates all of the working parts in the rocker box including the valve guides. The oil is either scavenged from the rocker box by means of a scavenge pump or is permitted to drain back to the crankcase through the push-rod housings. The cam drive gears are magnafluxed and inspected for wear and cracks. Some pitting has been experienced with these gears, but experience has indicated that this pitting can be corrected by reducing the unit loading through the addition of other gears. These parts will have a service life of upwards of 3000 hr.

The cam is magnafluxed and inspected for wear on the bearing and pitting on the cam track. The wear on the bearing is very low and the bearing will have a service life of over 4500 hr. Considerable trouble has been experienced on some engines with the cam track and roller follower becoming pitted and failing through fatigue. It is believed that it can be corrected by providing a uniform hardness gradient when the part is heat-treated. The cams in general use have a service life upwards of 3000 hr and are limited by their fatigue strength. The cam

rollers are in the same category, but have a service life of approximately 6000 hr.

The rate of wear on the tappets and guides is very low since they are well lubricated and have a low bearing loading. They will last approximately 9000 hr.

The push rods are checked for straightness and wear on the ends. The rocker arms are checked for cracks and condition of the bearings. These parts are limited in their service life only by fatigue strength if properly adjusted and have a service life of approximately 8500 hr.

The valve springs are magnafluxed and checked for cracks, wear, and loss of tension. The wear is very little and the springs usually are removed due to loss of tension after approximately 5000 hr. The loss in tension is no doubt a function of fatigue.

It would appear therefore that the limiting factor on the service life of the majority of these parts would be their endurance limit and that the factors which would govern the time between removals would be the condition of the cam track and rollers which in the average engine would be upwards of 3000 hr. There is a possibility of sludge accumulation jeopardizing the lubrication by clogging the passages but experience has not indicated this condition as serious during a 700-hr overhaul period. It would be open for question if the overhaul were extended to 3000 hr.

■ Accessory Drives

It is common practice for accessory drives to be provided with suitable mounting pads on the rear cover or the rear blower section for two magnetos, a starter, generator, vacuum pump, hydraulic pump, tachometer magneto, fuel pump, gun synchronizer, and an oil pump. These drives consist of the necessary spur gears and pinions supported by pressure-lubricated bronze, plain bushings with adequate oil seals to prevent external oil leakage. See Fig. 7. These parts are usually driven by an accessory driveshaft splined to the crankshaft and which, in some designs, also serves as a starter driveshaft.

The gears are splash-lubricated. The bushings are inspected for wear, which is very little and which limits the life of these parts to approximately 6000 hr of wear. These gears are relatively lightly loaded, and the rate of wear is low; for the same reason very few cracks are found. The endurance limit of the gears limits their service life to approximately 9000 hr.

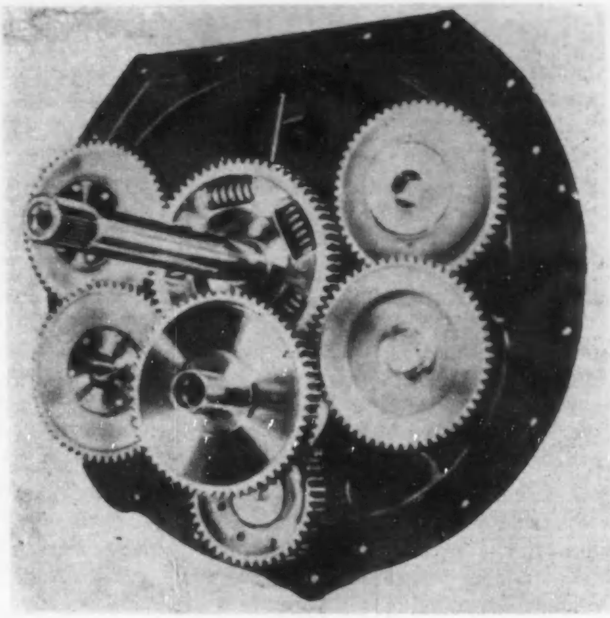
Some of these parts serve as passages in the lubrication system but, since the passages are large in cross-section area, they are not particularly vulnerable to sludge accumulation and deposits.

It is therefore assumed that this group of parts is not limited as to time between overhauls except by the wear on the bushing which would apply a limit of some 6000 hr.

■ Anti-Friction Bearings

It is current practice to support the crankshaft with either two or three roller bearings consisting of the conventional circular group of hardened rollers revolving between two circular races and with alignment maintained by a retainer. The thrust bearing is usually a ball bearing of conventional design although considerable success has been attained by use of the split-inner-race type bearing in this location.

All of the bearings are splash fed which, in some designs, is assisted by slinger rings of various types.



■ Fig. 7 – Front view of inside of Wright Cyclone GR-2600 rear cover with all accessory drives in place[5]

The bearings are inspected for wear, cracks, pits, chips, and cleanliness.

The service life of all of these bearings is limited by localized surface fatigue failures originated by surface damage due to foreign particles in the lubricating oil. This condition establishes the life on the roller bearings (main bearings) at approximately 2100 hr and at approximately 4000 hr on the thrust ball bearing. The loads are not uniform around the main bearing races due, of course, to the nature and direction of the acting forces, and it is general practice to rotate the bearings 120 deg each time they are reinstalled. However, it is believed that these parts would operate 1500 hr in one position if supplied with clean oil.

■ Oil Pump

The oil pump consists of two gear-type pumps operating in a common body and driven by one drive. The pump body also incorporates a spring-loaded relief bypass valve for pressure regulation. One of the pumps is a pressure pump which draws the oil from the oil tank and directs the oil under pressure to the relief valve and thence through the strainer into the system. The other pump is a scavenger pump and is usually approximately twice the capacity of the inlet pump. Its function is, as its name implies, to return the oil from the engine sump to the oil tank either directly or through the oil cooler as demanded by the temperature regulator.

The pump is inspected for wear of the gears, for cracks, and wear between the gears and the pump body. If there have been no metal particles going through the pump, the gears normally will be in good shape with no cracking and very little wear. Very little wear between the gears and the housing is experienced.

The oil pump is limited in service life by wear and its endurance limit. This life is considered to be approximately 8000 hr. The only limitation as to time between removals would be the accumulation of particles and sludge in the relief valve chamber which would continually be

[5] See "Double-Row Cyclone GR-2600 Engine," by E. G. Mason, Wright Aeronautical Corp. Service School.

Table 1 – Summary of Overhaul Period Limits on Various Groups of Parts

Section	Limit of Overhaul Period (Hr)	Due to	Most Important Factors That Affect This Limit				
			Fatigue	Wear	Lubrication	Deposits	Burning
1. Nose section, crankcase blower, and rear cover	3000	Fatigue of rear cover and deposits in oil passages	X			X	
2. Propeller shaft and reduction gear	1500	Pitting of teeth on stationary and reduction gear	X			X	
3. Crankshaft	1000	Crankpin deposits		X		X	
4. Master rod and link rods	1500	Wear and failure of bearing		X	X		
5. Cylinders and valves	3000	Cylinder-barrel wear and burning of valves		X	X		X
6. Pistons and rings	1500	Ring wear and carbon deposits		X	X	X	
7. Cam and cam train	3000	Condition of cam surface	X			X	
8. Accessory drives	6000	Bushing wear and gear-tooth failure	X	X			
9. Bearings (antifriction)	1500	Pitting of crankshaft main bearings	X		X		
10. Oil pump	8000	Gear and housing wear	X	X			
11. Oil pump relief valve	Periodic	Accumulation of deposits			X	X	
12. Oil strainer	Periodic	Accumulation of sludge and deposits			X	X	

sticking the relief valve. This part is removable and can be cleaned periodically in the field without removal of the pump.

Oil Strainers

Some engines incorporate a mesh wire screen and others a rotatable disc type strainer both of which are provided with suitable bypass facilities. It is current practice to clean these strainers periodically and, since they are removable without disturbing related parts, they are not to be considered a limiting factor to the overhaul period.

Discussion

Table 1 is a tabulated summary of the engine inspection and expected findings just discussed. This summary lists the maximum time between overhauls for the particular group of parts, a description of the cause for the limit, and denotes the factors which are considered most important to the overhaul period and life of these parts. The causes listed are not intended to imply that these are the only pertinent factors that affect the life of the particular part but instead these are believed to be the most important factors. The writer again wishes to call attention to the previously stated condition that each of these limiting factors may not necessarily be true for any one given model engine in current use but, consistent with the best available information, the factors listed are applicable to our hypothetical engine which is intended to be representative of the currently used engines as a group.

It is obvious from Table 1 that, in the writer's opinion, the mechanical limiting factor for overhaul periods is the crankpin deposit and the general lack of cleanliness of the engine and the oil. Also, that the factors most predominantly affecting the service lives of the majority of the other engine parts are wear and lack of proper lubrication, which is intended to mean the quality of the oil and the distribution of clean oil. Wear and lack of proper lubrication are inseparable but, since it is generally conceded that the latter precedes the former, we will refer to the combination as lack of the proper lubrication.

No doubt, many operators will question as to why burned valves and sticking valves are not listed as limiting factors since these conditions have been a most prolific source of malfunctioning on several types of engines during the last few years. The cause for these conditions is subject to debate but the indications are that the cause is not necessarily a function of time. To substantiate this belief, Fig. 8 is presented which shows the time since overhaul of 437 cylinders removed during the months of August, September, and October, 1941, by one airline due to known cases of burned valves. Fig. 9 is also presented showing the distribution of these cases of burned valves around the engine.

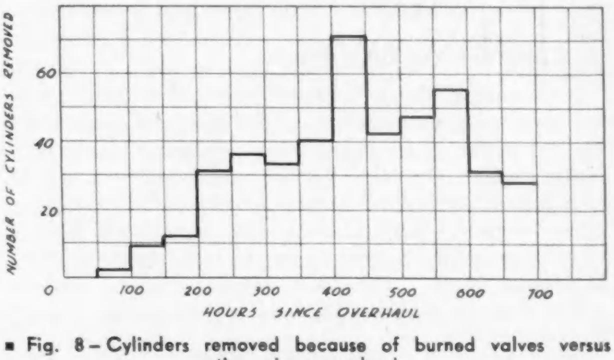
It is believed that burned valves are due to improper cylinder-head design in regard to heat dissipation, possibly coupled with poor heat-transfer characteristics of some pressed-in valve seats. These conditions might be augmented by an unequal mixture distribution between cylinders or by slight irregularities in the intercylinder baffles causing irregular cooling. This opinion is based on observations which indicated the valve seats to be warped in a consistent manner.

Sticking of valves is in the same general category except possibly not quite as distinct. The writer feels that the valve-guide designer has not recognized the physical limitations of the available oil and, in this location, is expecting the oil to do an impossible job. However, the fact that these discrepancies are not considered as limiting factors does not relieve the manufacturers of the responsibilities of correcting the conditions.

There are a number of other deficiencies that cause chronic difficulties in service such as the ignition harness and spark plugs which are in the same category as the valves and valve guides in regard to their effect on the time between overhauls. Each of these parts is a study within itself and, to be consistent with the intent and scope of this paper, it is considered advisable to discuss only the factors which are considered the limiting factors; namely, the crankpin deposit and general lack of cleanliness of the engine.

The importance of the crankpin deposit is that it not only throws the carefully balanced crankshaft out of balance but, as the deposit level reaches the end of the oil tube, the foreign material normally centrifuged out is allowed to pass into the master-rod bearing where it promotes wear. As previously mentioned, this bearing is very vulnerable to wear because of its function in the lubrication system. This fact is particularly important in those engines depending solely on bearing clearances for controlling the flow of oil into the power section of the engine.

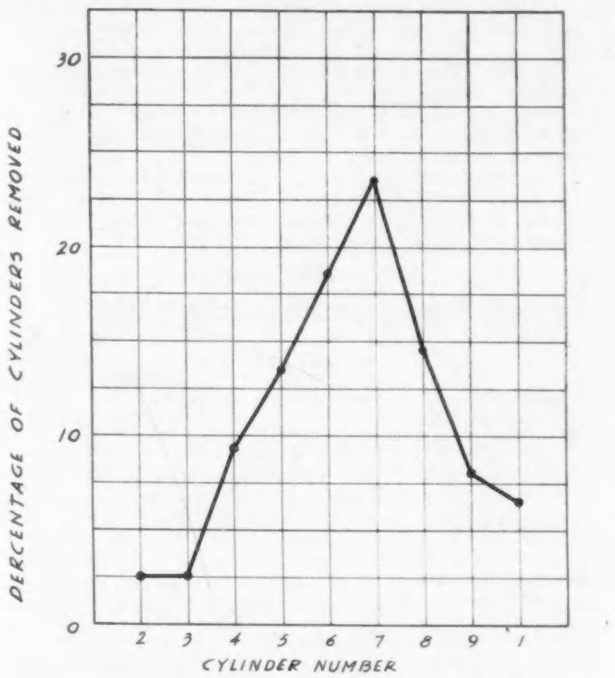
The cleanliness of the engine and of the oil is, of course, important because of its effect on wear of the various parts,



■ Fig. 8 – Cylinders removed because of burned valves versus time since overhaul

on propeller synchronization, on oil flow through various passages, and on the operation of variable-speed supercharger-drive mechanisms.

By lack of proper lubrication, the writer does not intend to imply that the quality of the oil is solely at fault but that the failure to provide a properly distributed supply of clean lubricating oil is equally deficient. The designer of the lubrication system is to be criticized for his failure to recognize the physical limitations of currently available lubricating oils and for not designing accordingly. The writer does not condone the tendency for present oils to sludge, to foam, to cause ring gumming, to fail to protect working parts from corrosion or to corrode special alloy bearings; nor is the oil refiner relieved of his responsibility of further improving the oils. However, it is contended that closer coordination between the oil refiner and the engine manufacturer and cognizance of their mutual limitations should be reflected in engine designs to a greater degree than at present.



■ **Fig. 9 – Location of cylinders removed because of burned valves**

In view of the papers previously written on aircraft-engine lubricating oils in which the function and chemical aspects of oil have been so ably discussed and in view of the contentions just stated, this paper will deal only with the mechanical and physical aspects of the lubrication system that are pertinent to the purpose of this paper.

An analysis of a typical sample of this crankpin deposit is as follows[6]:

	%	Source
Soluble in ASTM Naphtha	8.01	Mineral oil present as such -- from oil
Insoluble in ASTM Naphtha Soluble in Chlorine	0.75	Gums and resins – from oil
Carbon and Carbonaceous Matter	2.97	From oil
Silica and Silicious Matter	1.09	Sand and dust contamination
Lead as Pb	69.41	From fuel
Iron as Fe_2O_3	3.39	From wearing parts
Halides as Bromine	15.6	From fuel
Specific Gravity at 80 F		4.159
Weight, oz		12.5
Time Since Overhaul, hr		538

It is noted that a large percentage of this deposit is lead oxide from the fuel; however, it must be remembered that this sample is a centrifuged deposit and the percentages shown are the ratios in which the constituents are centrifuged and deposited which in turn is dependent on their respective gravities. These percentages, therefore, do not indicate the relative percentages of the materials as they exist in the oil.

Lead oxide, as such, may not seriously affect wear; however, it does offer a generous source of material for filling the crankpin cavity and for clogging passages during extended overhaul periods, thereby jeopardizing the lubrication system. Also it is a parasite to a centrifugal-type filter requiring a larger filter to do a job that could be accomplished by a smaller unit if the lead oxide was not present. Therefore, the first requirement for improved lubrication facilities is to decrease the amount of lead oxide in the lubricating system because of its "parasitic" or "nuisance" value. This condition can be alleviated by improving the piston-ring sealing to reduce blowby and by finding means of more rapid scavenging of the lead products formed during combustion. Therefore, the correction of this deficiency becomes the dual responsibility of the engine manufacturer and of the Ethyl Gasoline Corp.

The removal of the other contaminations indicated in the analysis requires facilities for effective filtering of the lubricating oil. As previously explained, the current engines direct the oil under pressure through a strainer into passages, and approximately half-way through its course an attempt is made to centrifuge the oil using a centrifugal effect in a chamber which happened to be there because the designer wanted to lighten the crankshaft. Fortunately, the crankpin has done a fair job at the sacrifice of crankshaft unbalance but its capacity dictates the frequency with which it must be cleaned.

The strainers now used sometimes referred to as filters do a fair job of collecting nuts and bolts or at least they do a fair job of accumulating such foreign matter until enough is present to restrict the flow and cause the bypass to open at which time the whole collection is dumped into the "filtered oil" passages. In fact, the disc-type filter is a very "democratic" unit – if it happens to stop a thin carbon flake, which is the normal shape of these particles, because the flake hit the filter broadside, the wiper blades dislodge the particle, giving it another chance. This procedure continues until the particle either hits the disc stuck sidewise and passes through or until enough particles are collected to cause the bypass valve to open and permit dumping them all into the system at one time.

Additional filtering efficiency could be attained by the incorporation of one of the two conventional-type filters; namely: (1) the filter-pack type which depends on the flow velocity being reduced and the oil being passed through restricted passages formed by the pack material, thereby mechanically stopping the foreign particles; or (2) the whirling-disc, centrifugal-type filter either engine driven or oil scavenge pressure driven which utilizes the effect of the centrifugal force on gravity of the particles. Both types of filters are, of course, subject to criticism due to the added weight, size, and power required to drive them, as in the case of the centrifugal type. Also they necessarily have to be bypassed during cold starting, thereby subjecting the lubrication system to unfiltered oil.

From the standpoint of weight, it is simply a problem

[6] See American Airlines Engineering Report No. 299, by G. K. Brower.

of the value of the weight to the operator versus the reduction in overhaul and maintenance costs attendant to the use of such a filter.

It has been demonstrated on a 100-hr test-stand run[7] that a centrifugal-type filter, 5½-in. inside diameter by 3¼-in. length turning 4000 rpm is some 22.5 times as efficient as the crankpin having removed 946.6 g of naphtha insolubles as against 42 g removed by the crankpin[8]. This filter, known as an "airfuge," was constructed as a separate unit to be installed on the engine firewall and driven by the scavenge pump oil pressure. The total installed weight of the experimental unit including lines, brackets, additional oil, and so on, is approximately 23 lb; however, this weight could be reduced appreciably, possibly to one-half of the present weight if incorporated in the engines.

Analysis of the centrifuged samples from this test indicated that the crankpin sample contained a greater percentage of lead than did the centrifugal filter sample. The manufacturer of the unit interprets this finding to mean that the crankpin is selective in its clarification to the high-gravity constituents and estimates that the centrifugal-type filter of this size is 40 to 50 times as efficient as the crankpin in removing material damaging to the engine.

The lack of factual data prevents a comparison of the two types of filters; however, the filter-pack type unit is inherently heavy due to its size which is required in order to reduce the flow velocity.

Another frequent deficiency in some engine lubrication systems is the custom of placing a spring-loaded ball or plunger type pressure-regulating bypass valve between the pump and the strainer instead of on the "clean-oil" side of the strainer. This is done, of course, to save weight and to reduce the size of the strainer; however, this weight saving is paid for at a terrific price in loss of oil pressure due to binding of the relief valve resulting in momentary starvation of lubrication and in operating difficulties. This starvation of lubrication is obviously not conducive to low wear rates in the engine.

There is also a very dire need for a correction of the oil foaming phenomenon which has been current for a number of years and which jeopardizes the continuity of parts lubrication. When this foaming occurs in flight, oil foam is emitted from the engine breather or from the oil tank vents or cap with great rapidity and proceeds to cover the engine nacelle and wing center section as well as the side of the cabin including the windows. The passengers' reaction is even worse than if the engine had failed mechanically because there is more visible evidence present.

The cause for the foaming is a much debated subject; however, the condition would be greatly alleviated by providing facilities to remove mechanically the entrapped air from the scavenged oil before it reaches the oil tank. Air is separated from the steam in the parlor radiator, and air is removed from the gasoline in an ordinary refueling truck. Why can't an aircraft engine be provided with similar facilities?

Assuming that these deficiencies were removed or sufficiently alleviated, Table 1 indicates that the overhaul periods could possibly be extended to 1500 hr. At this point, the limiting factors would be:

1. Pitting of the reduction-gear teeth. 2. Wear and failure of master-rod bearing. 3. Ring wear and carbon deposits in the ring grooves. 4. Pitting and fatigue of the crankshaft main bearings.

The severity of all of these conditions, particularly Items (2) and (4), would be lessened by the correction of the lubrication deficiencies previously described. Item (1) could be further improved by providing a controlled hardness gradient during the heat-treating process and plating of piston rings, plating of cylinder barrel surfaces, and improvement in materials offer possible benefits to (3).

■ Summary

The limiting factor for the optimum or desirable overhaul period may be an economic factor due to the inability to utilize all of the life of the various parts on which the initial cost write-off per hour is high.

The limiting factor to the maximum overhaul period is concluded to be the volume of the crankpin centrifuged deposit and general lack of cleanliness of the engine. These factors can be alleviated by decreasing the tendency for the oil to form sludge, by decreasing the amount of lead in the lubrication system through improvement in ring sealing and fuel refining, and by providing effective filtering facilities in the lubrication system.

Should this factor be eliminated successfully, the overhaul period would be expected to be limited at 1500 hr by:

1. Pitting of the reduction gear teeth. 2. Wear and failure of the master rod bearing. 3. Ring wear and carbon deposits in the ring grooves. 4. Wear on the crankshaft main bearings.

It is also pointed out that there is a definite need for a relocation of the oil pressure regulating valve, for facilities to remove the entrapped air from the oil to alleviate the foaming condition, for a correction of the valve burning and sticking condition, and for an improved ignition harness.

The writer wishes to express his appreciation to the Maintenance and Engineering Departments of Chicago and Southern Airlines, Eastern Air Lines, United Air Lines, and American Airlines for their invaluable assistance in the preparation of this paper.

DISCUSSION

Favorable Experience With Centrifugal Filter

– E. A. Ryder

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MR. WHITLOCK, in his interesting paper, speaks of the "Average Engine." Of course, there is hardly such a thing as an average engine any more than there is an average man. Mr. Whitlock puts it up to the engine builders to provide a more durable engine than he says is common at present but, in lumping the good and bad together in order to get average figures for the service life of different parts, he ignores those engine models which are giving a better-than-average performance. Eliminating the comments on mechanical troubles which are peculiar to certain current engine models, but which definitely can be eliminated by changes in design or material, the paper is really a strong plea for filters in the oil system. Pitting of reduction-gear teeth, wear and failure of the master-rod bearing, and wear of the crankshaft main bearings can be and have been eliminated or greatly postponed on many engine models that are suitable for transport operations. This leaves piston-ring sticking and plugging of oil cavities as the items which will limit the length of overhaul periods.

Sludge may collect in a crankpin cavity or the inside of a reduction-gear pinion axle up to the level of the oil outlet hole. Beyond this point, however, it is our experience that the oil keeps a channel open, and actual plugging does not occur. With unsuitable bearing materials, it is possible that the passage of sludge through the bearing clearance may cause undue wear. This is not true of all bearings.

It has been our experience and that of others that the use of a centrifuge in the oil system collects enough solid matter from the oil so that the filling of sludge cavities proceeds at a slower rate than otherwise. Collection of sludge in the crankpin cavity seems to proceed at about one-third to one-half the normal rate when a centrifuge is used. This experience suggests that the overhaul period might be lengthened very materially by the use of a centrifuge or possibly even by a stationary filter.

To avoid the weight and cost of a full-flow filter a small bypass filter might be used if it were so located that it could be easily serviced at rather frequent intervals, say every 25 to 50 hr. Very little time would be required to open the filter chamber and replace the filtering medium.

In his introduction Mr. Whitlock criticizes the engine manufacturer for increasing engine power. I think it is obvious that the engine builder would be only too glad to have the user de-rate the engine, thereby gaining greater reliability and lower operating costs. It is the economics of airline operation that prevents this, and not the skullduggery of the engine builders.

[7] See Wright Aeronautical Corp. Report No. 491, July 2, 1940.
[8] See Technical Department Report No. 32A-JJS-22, The Sharples Corp., by J. J. Serrell.

Effect of Diesel Fuel on Exhaust Smoke and Odor

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and

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THIS paper reports the results of an extensive program of laboratory investigation of diesel exhaust smoke and odor carried out on two popular automotive-type diesel engines. Highlights follow:

Although fuels may be chosen to give desired results along any selected lines of engine performance, it is not possible to obtain all desirable results with a single fuel, at least not without recourse to additives.

The only present solution appears to be fuel selection on the basis of a compromise amongst the various factors involved. Proper interpretation of the engine performance characteristics most desired and consideration to the type and mechanical condition of the engine used should result in selection of fuels for utmost satisfaction.

Although pronounced exhaust smoke and odor occur under full load, the most objectionable conditions were those occurring during idling, acceleration after idling and, occasionally, deceleration with the clutch engaged. From this finding and the results of the tests made, it appears that an increase in cetane number will produce a decided decrease in exhaust smoke by virtue of its ability to support combustion under adverse conditions of low output.

In an attempt to minimize the obnoxious smoke and odor from exhaust gases of diesel-powered vehicles, numerous operators have resorted to kerosene-type fuels. While this expedient did result in a substantial reduction in exhaust smoke, it was accompanied by a pronounced loss in power and economy due to the lower available heating value on a volume basis.

THE increase in the number of high-speed diesel engines used in truck and bus service in the United States has been extremely rapid during the past decade. With the large number of such vehicles used in transportation, particularly in metropolitan areas, considerable difficulty has been experienced with obnoxious smoke and odor from the exhaust gases. In an attempt to alleviate this condition numerous operators resorted to kerosene-type fuels. While this expedient resulted in a substantial reduction in exhaust smoke as pointed out by Manville and associates[1], it was accompanied by a pronounced loss in power and economy due to the lower available heating value on a volume basis.

In view of the foregoing shortcomings of both "normal" and kerosene-type fuels, it became increasingly important to have available exact information as to the influence of individual fuel properties on engine performance as reflected by exhaust smoke and odor, power, and economy. To this end an extensive program of laboratory investigation was carried out in two popular automotive-type diesel engines.

Preliminary work on the subject consisted of a critical survey of several fleet operations and field tests conducted on a number of buses. The results of this work confirmed reports that, although pronounced exhaust smoke and odor occur under full load, the most objectionable conditions were those occurring during idling, acceleration after idling, and occasionally deceleration with the clutch engaged. Smoke and odor occurring during the latter condition are due to governor maladjustment which permits fuel injection with the engine acting as a brake.

Method of Test

Most of the work herein described was conducted on Engine A; this work was later supplemented by limited tests on Engine B of approximately the same size but

[This paper was presented at the 15th ASME National Oil and Gas Power Conference, the SAE Diesel Engine Activity Cooperating, Peoria, Ill., June 19, 1942.]

[1] See SAE Transactions, Vol. 35, October, 1940, pp. 397-408: "The Control of Smoke in the Automotive Diesel," by W. W. Manville, G. H. Cloud, A. J. Blackwood, and W. J. Sweeney.

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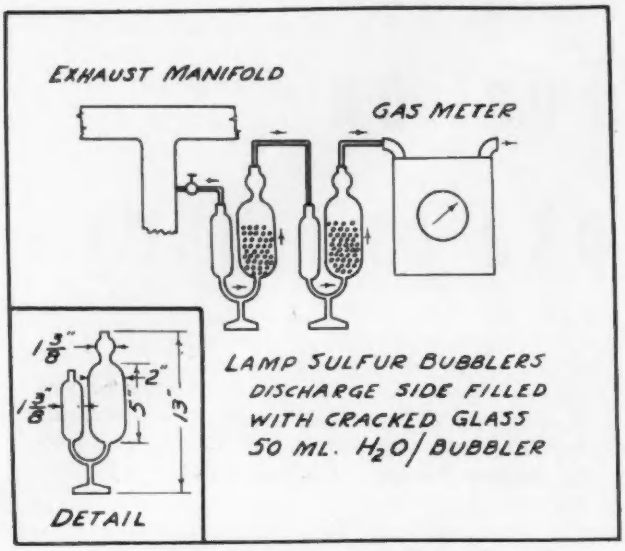
THE AUTHORS: R. S. WETMILLER worked as research assistant in mechanical engineering for two years at the Pennsylvania State College after receiving his M. S. degree in mechanical engineering from this college in 1935. Since then he has been engaged in fuels and lubricants work with The Texas Co., having become assistant supervisor of engineering research in 1940. CAPT. L. E. ENDSLEY, JR., received his M. S. degree in mechanical engineering from Purdue University in 1936. From that time up until 1939 he conducted engine work in The Texas Co.'s research laboratory, when he went with the Dayton laboratory of the Army Air Corps on a leave of absence. He returned to The Texas Co. in 1940 to continue his engineering research work, and was called into the service as first lieutenant early this year.



■ Fig. 1 – Exhaust-gas sampling apparatus

possessing radically different design features. The observations taken on both engines included measurement of smoke and odor of the exhaust gas and power and economy under a wide variety of operating conditions.

The smokemeter used consisted of an adaptation of that used by MacGregor[2] which measures the opacity of the exhaust gas in the exhaust pipe itself. This instrument was chosen in preference to the sampling type since it is better adapted to measurement of very dense smoke and to rapid observations such as during acceleration. The odor measurement of the exhaust gas was by two methods, olfactory rating by a number of observers, and a more convenient method consisting of chemical analysis of the exhaust gas to determine its aldehyde content in terms of formaldehyde. The method of aldehyde determination has been described in detail by Mikita, Levin and Kichline.[3] In evaluating a large number of fuels it is convenient to supplement smell ratings by the absolute basis of comparison made possible by the chemical method. Accordingly, odor results are reported as milligrams of aldehydes, in terms of formaldehyde, per cubic foot of exhaust gas. Relative odor ratings were also obtained on each group of samples tested; however, these ratings are not comparable from group to group due to the absence of an absolute scale of measurement. Power and economy results were obtained in the conventional manner by the use of a d-c electric dynamometer and a volume measure of the fuel quantity consumed.

The method of exhaust-gas sampling for aldehyde determinations described by the foregoing authors[3] was modified slightly during the course of the present work to allow for convenience of operation and better precision. Modification consisted of substituting two glass lamp sulfur bubblers in series for the condensate traps previously employed. Fifty ml of distilled water were used for aldehyde absorption in each bubbler and the composite sample used for aldehyde determination. Three cu ft of exhaust gas were passed through the apparatus at a rate of 0.5 cfm for each determination. A diagrammatic sketch of the apparatus is shown in Fig. 1. During the development of this method three rather than two bubblers were used, but the third was eliminated since it consistently collected no aldehydes.

The engines were maintained in first class mechanical condition during the entire project, particularly with re-

[2] See SAE Transactions, Vol. 31, May, 1936, pp. 217-224: "Diesel Fuels – Significance of Ignition Characteristics," by J. R. MacGregor.

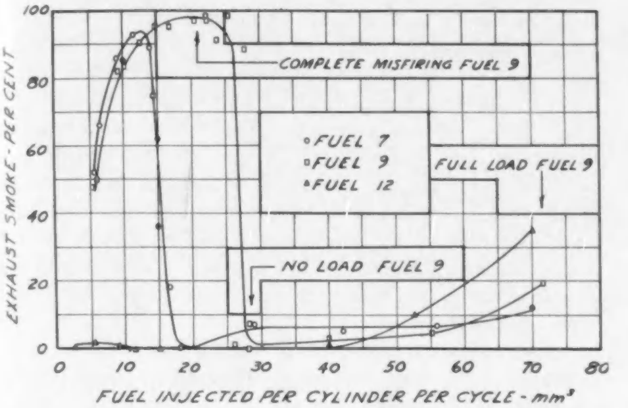
[3] "Gasoline Engine Exhaust Odors," by J. J. Mikita, Harry Levin, and H. R. Kichline, presented at the National Fuels and Lubricants Meeting of the Society, Tulsa, Okla., Oct. 7-8, 1942; scheduled for publication in SAE Transactions. Their method of exhaust gas sampling and analysis is given in Appendix I.

Table 1 – Inspection Tests on Fuels Used

Fuel Number	API Gravity, deg	Cetane Number	S. U. Viscosity* at 100 F, sec	ASTM Distillation, F 10%	50%	90%
			Commercial-Type Fuels			
1	35.4	47	35.1	451	502	556
2	43.0	50	31.3	388	417	458
3	35.2	44	36.0	460	504	558
4	43.7	57	33.8	410	477	604
5	43.3	43	31.2	387	416	462
6	23.1	38	73.0	573	621	719
7	37.5	56	37.6	470	523	582
8	26.7	40	51.0	528	600	678
9	29.0	33	37.6	440	500	573
10	40.8	44	31.3	391	412	451
11	32.5	48	41.3	500	547	650
12	44.0	72	35.1	401	493	605
13	21.3	20	35.1	428	480	584
			AB-PB Blends			
14	32.6	27	33.2	386	419	506
15	33.1	31	33.0	395	428	512
16	37.4	41	32.3	394	426	505
17	42.2	52	31.7	395	430	502
18	30.5	27	35.0	410	453	519
19	31.1	31	34.5	423	465	526
20	35.4	39	33.9	425	463	524
21	40.4	51	33.2	422	457	516
22	44.8	64	32.5	420	456	414
23	28.4	29	37.0	450	487	534
24	29.1	31	36.9	457	495	539
25	32.6	41	36.4	464	498	544
26	36.8	51	35.9	469	499	544
27	40.9	65	35.3	475	502	550
28	24.9	34	51.6	546	565	609
29	27.2	42	49.2	557	579	622
30	31.1	52	46.5	560	582	626
31	35.2	62	44.2	560	584	630
70	48.2	60	31.2	386	415	484
71	46.3	64	32.2	414	448	506
72	43.7	73	35.0	470	497	543
73	40.0	71	42.3	554	573	614
			LC-PB Blends			
32	26.2	21	31.4	398	425	476
33	30.2	31	31.4	398	425	480
34	35.5	42	31.4	396	423	482
35	41.2	51	31.2	395	424	489
36	23.2	22	32.3	422	446	496
37	27.3	29	32.3	422	447	497
38	32.3	39	32.2	423	450	504
39	38.3	51	32.2	423	450	505
40	44.5	62	32.1	423	455	511
41	19.6	22	34.8	468	490	539
42	23.8	30	34.8	470	491	536
43	28.5	39	34.8	473	493	538
44	34.5	50	34.8	473	498	545
45	39.6	63	34.8	475	499	546
46	16.5	25	46.0	543	573	636
47	19.7	31	45.3	539	575	644
48	23.3	40	44.8	544	578	644
49	28.6	46	43.9	550	584	641
50	33.2	59	43.2	558	585	639
70	48.2	60	31.2	386	415	484
71	46.3	64	32.2	414	448	506
72	43.7	73	35.0	470	497	543
73	40.0	71	42.3	554	573	614
			HA-PB Blends			
51	51.1	21	33.5	401	417	462
52	50.3	33	32.7	398	419	470
53	49.5	42	32.0	396	422	477
54	48.7	50	31.5	390	418	483
55	49.3	21	35.8	423	446	488
56	48.3	34	34.6	425	452	497
57	47.7	42	33.8	423	452	507
58	47.1	54	33.0	421	450	504
59	46.4	63	32.3	415	449	508
60	45.7	21	41.2	470	496	540
61	45.4	30	39.3	473	495	542
62	44.9	43	37.4	471	496	542
63	44.4	51	36.4	470	496	542
64	43.8	65	35.2	473	497	544
65	41.6	20	69.4	538	575	645
66	41.3	30	63.5	543	578	642
67	40.7	41	52.9	551	583	643
68	40.2	51	45.2	557	586	639
69	39.9	66	41.0	562	588	638
70	48.2	60	31.2	386	415	484
71	46.3	64	32.2	414	448	506
72	43.7	73	35.0	470	497	543
73	40.0	71	42.3	554	573	614

* Converted from kinematic.

spect to injector repair. At the start of each day's operation the engine was checked for performance on a reference fuel of known behavior and, when necessary, suitable adjustments were made to duplicate previous results. In most cases these adjustments consisted of injector reconditioning; fuel evaluations were found to be very critical to injector condition. Unless otherwise noted, manufacturers'



■ Fig. 2 – Effect of fuel injection rate on exhaust smoke – Engine A – 1400 rpm

recommended settings were used throughout. All tests were conducted with a jacket temperature of 170 F and crankcase-oil temperature of 180-200 F. SAE 30 oil was used throughout with oil changes at 30-40 hr of operation.

Due to the interrelation between various fuel properties, it is possible to show some type of agreement between engine performance and most physical properties of the fuels. However, results herein described have been related to the various fuel properties which are believed to be directly responsible for the effects observed.

■ Causes of Exhaust Smoke

The 13 commercial-type fuels shown in Table 1 were used in the initial examination to trace the course of exhaust smoke and odor formation. These tests were conducted in Engine A under equilibrium conditions of operation from full load to motoring with fuel injection; the latter condition simulates governor maladjustment which permits fuel injection with the engine acting as a brake. Particular emphasis was placed on low-output conditions as suggested by observation of fleet performance.

It was observed that, as engine output (fuel rate) is decreased from high load conditions, exhaust smoke decreases until a point is reached where there is a sudden increase in smoke. Further decrease in fuel rate results in increased smoke until a maximum is reached after which smoke decreases when combustion ceases. Fig. 2 demonstrates this behavior for three typical fuels under equilibrium conditions.

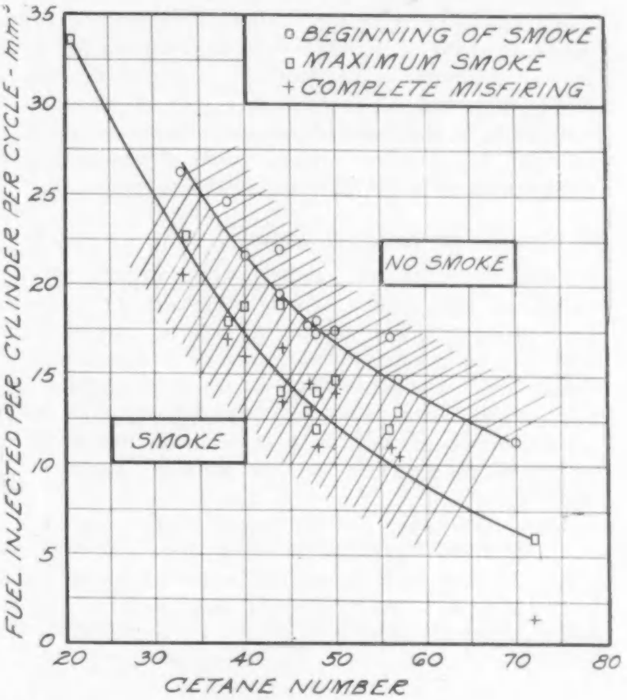
Examination of fuel sprays in bench tests disclosed that, as injection rate is decreased, the spray form deteriorates until at very low rates only dribbling is experienced. This behavior, together with lean mixtures, appears to explain the engine performance observed.

It was observed further that smoke occurring at fuel rates above that for minimum smoke was predominantly of the black type resulting from unburned carbon particles in the exhaust gas as evidenced by soot collected on cotton-cloth exhaust targets. Below this value, smoke was predominantly of the white type resulting from vaporized unburned fuel; exhaust targets obtained in this region consistently showed fuel droplets and freedom from soot. It was noted further that, in all cases where white smoke was obtained, liquid fuel was found present in the exhaust system.

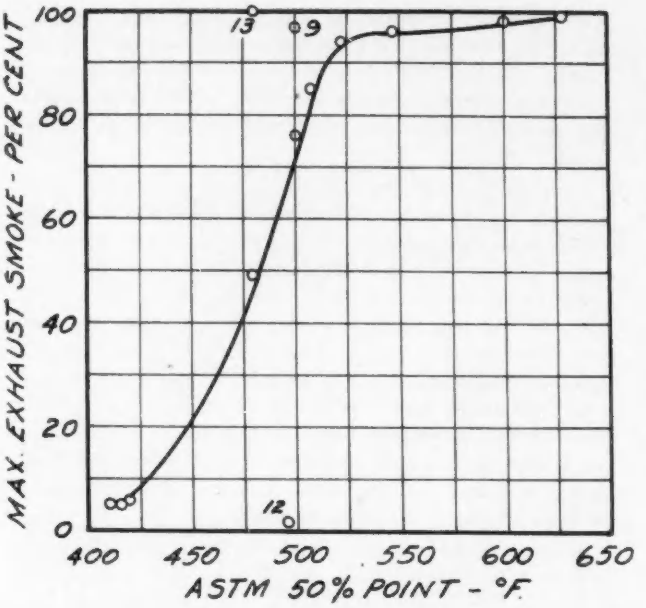
Both the fuel rate for sudden increase in smoke and the fuel rate for maximum smoke depend upon cetane number, as shown in Fig. 3 for the 13 fuels tested. Maximum smoke occurs at a fuel rate approximating that for complete misfiring as evidenced by the motoring horsepower being equal to the friction horsepower. An increase in cetane thus affects horizontal displacement to the left, the low injection rate portions of the curves in Fig. 2. Since poor spray form and lean air-fuel ratio, such as exist under these conditions, are conducive to increased ignition lag, low cetane fuels ignite very late in the cycle; with high cetane, ignition is sufficiently early to allow reasonably good combustion and reduced smoke.

The magnitude of maximum smoke is apparently related to volatility for fuels of 38 to 57 cetane, as shown in Fig. 4. The points that lie off the curve suggest that more complete data would result in a family of curves with lower cetane fuels showing more smoke for a given volatility.

Although the operating conditions outlined in the foregoing tests do not, as a rule, occur in service, the results are believed to be directly applicable to field performance. If conditions were such that liquid fuel collected in the exhaust system, any subsequent operation at high exhaust temperatures would be expected to result in rapid vaporization of the collected fuel and dense exhaust smoke. In the case of very high cetane fuels, combustion would be substantially complete even under very adverse conditions, hence, little if any liquid fuel would be collected in the exhaust system and a low order of exhaust smoke should be observed. Any decrease in the cetane number would



■ Fig. 3 – Effect of cetane number on exhaust smoke – Engine A – 1400 rpm



■ Fig. 4 – Effect of volatility on exhaust smoke – Engine A – 1400 rpm

result in less complete combustion with more liquid fuel collected and increased exhaust smoke. With the more volatile fuels, even though combustion were incomplete, less liquid fuel would be collected in the exhaust system during misfiring due to the temperatures of the system being sufficiently high to prevent appreciable condensation. Again, a low order of exhaust smoke would be expected.

The foregoing deductions were verified for the entire series of 13 fuels by operating under conditions of complete misfiring followed by full-load operation and by idling followed by ¾ load operation. Results of these runs are shown in Appendix II.

From the foregoing work, it appears that an increase in cetane number will produce a decided decrease in exhaust smoke by virtue of its ability to support combustion under adverse conditions of low output. However, if a fuel is subjected to conditions such that complete combustion will not occur due to its low cetane number, an increase in volatility will decrease exhaust smoke through reducing the quantity of liquid fuel in the exhaust system.

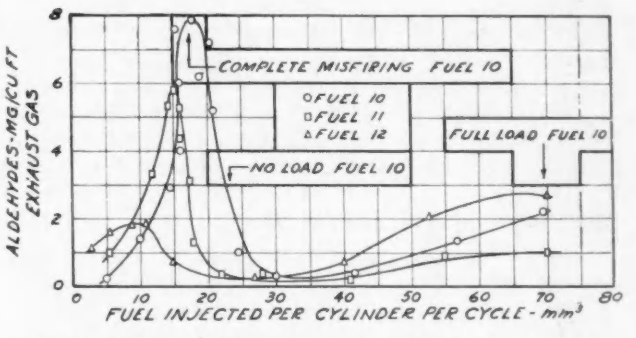
■ Causes of Exhaust Odor

Paralleling experience with gasoline engines,[3] it was found that the pungent odor of diesel exhaust gas is related to its aldehyde content in terms of formaldehyde. This fact was confirmed on numerous fuels and under a variety of operating conditions by smell ratings and aldehyde determinations on exhaust-gas samples taken simultaneously. Data to be presented later were obtained by both methods and will serve to indicate the relationship obtained.

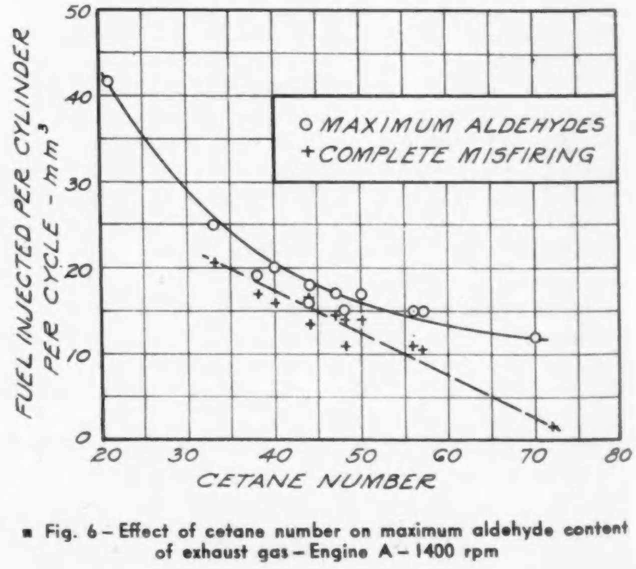
It was determined that, as engine output (fuel rate) is decreased from high load conditions, exhaust odor decreases until a point is reached where there is a substantial increase in odor. Further decrease in fuel rate results in increased odor until a maximum is reached at the lean limit of combustion. Typical results are shown for three fuels in Fig. 5 which is a plot of exhaust odor in terms of aldehydes obtained at constant speed under equilibrium conditions. The similarity between smoke and odor results as demonstrated by Figs. 2 and 5 is noteworthy and suggests that low-output conditions favoring smoke also favor formation of the most objectionable odors.

Although differences between fuels with respect to exhaust odor apparently exist at high load conditions, the magnitude of aldehydes obtained and differences between fuels are of minor importance when compared with those at low load conditions. This conclusion was verified by odor ratings on exhaust-gas samples over the entire range from motoring to full load. Two distinct types of odor were distinguished, one from rich and one from lean operation as was the case in gasoline engines.[3] Rich odors were strong and heavy, but not particularly objectionable while lean odors were very pungent, sharp, acrid and objectionable. In most diesel engines the air charge per cylinder per cycle is substantially constant at constant speed, hence rich odors are encountered at and near full-load operation under conditions where an excess of fuel is injected as evidenced by black smoke. Lean odors are encountered in the white smoke region at low loads and grow increasingly intense as the load is decreased. At part-load operation odors are of such a low order as to be negligible. In this region combustion is substantially complete as evidenced by an almost complete absence of smoke.

In the lean region, high-cetane fuels tend to reduce the fuel rate for maximum aldehydes, as shown in Fig. 6, for the 13 fuels examined. Maximum aldehydes occur just before the lean limit of combustion is reached. Again an



■ Fig. 5 – Effect of fuel injection rate on aldehyde content of exhaust gas – Engine A – 1400 rpm



■ Fig. 6 – Effect of cetane number on maximum aldehyde content of exhaust gas – Engine A – 1400 rpm

analogy is noted between smoke and odor performance exemplified in Figs. 3 and 6.

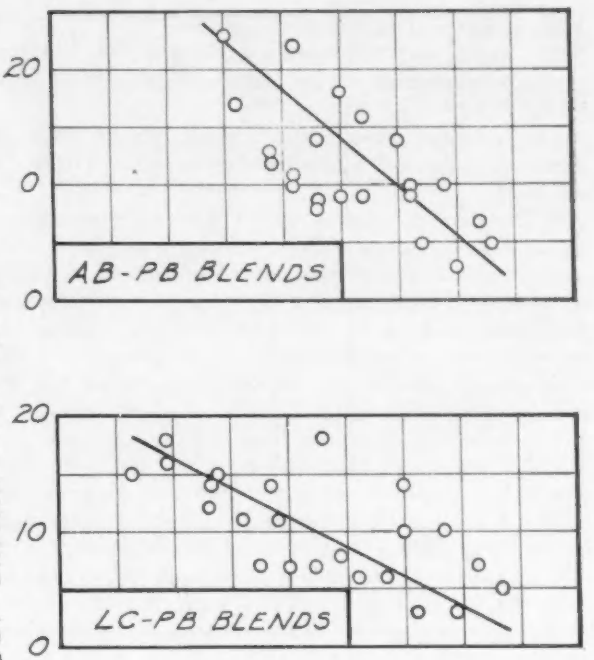
Under the operating conditions employed, it is indicated that exhaust gas odor depends only upon completeness of combustion in the cylinder and is independent of the presence of liquid fuel in the exhaust system. It is therefore evident that odor depends only on cetane number which is a reflection of the ability of the fuel to support combustion. It might be expected that, if conditions were such that liquid fuel collected in the exhaust system, subsequent operation at high exhaust temperatures would result in rapid vaporization of the collected fuel and the formation of objectionable odors. If this condition were true, volatility should have some influence on exhaust-gas odor as was the case with exhaust smoke. However, inasmuch as odor is a function of volume of exhaust gas discharged per unit time, it is doubtful if the small amount of gas originating from evaporation of the liquid fuel would be sufficient to produce additional objectionability to the total volume of discharged gas. Although this point was not exhaustively investigated, such data as are available confirm this line of reasoning.

Performance of Special Fuels

Special fuels used to determine the influence of individual fuel properties on engine performance were made to be as identical as possible in all but one of the following variables which covered a much wider range than most commercial products: fuel type, cetane number, and volatility. These were prepared by cutting the following stocks and blending with paraffin base (PB) gas oil in all cases in order to obtain the desired spread in cetane number:

1. Naphthene base (AB) gas oil
2. Light cycle (LC) gas oil
3. Heavy alkylate (HA)

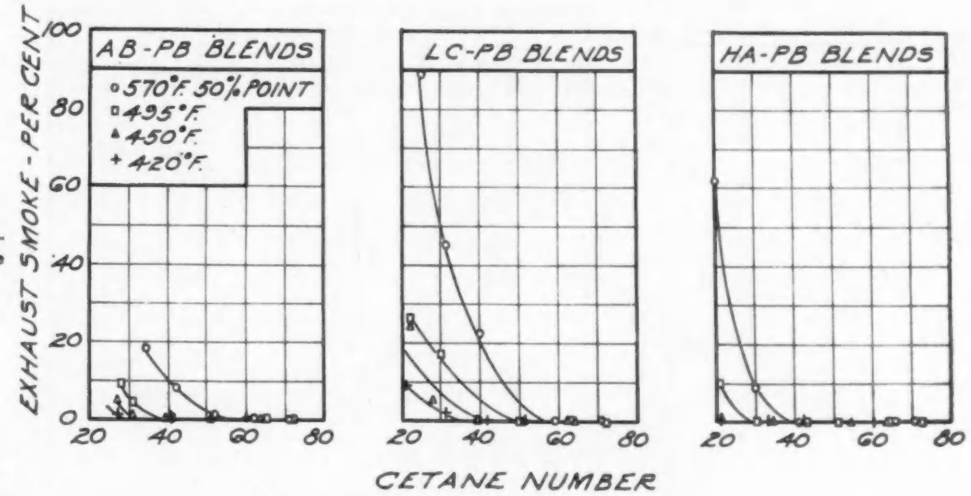
In the lower cetane range this procedure furnished fuels representing predominantly naphthenic, unsaturated cyclic and isoparaffinic stocks, respectively. In the higher cetane region all blends were predominantly normal paraffins because of the large amount of PB blending stock required in these cases. The desired spread of 20 to 70 cetane number was obtained in most cases in each volatility range represented by ASTM 50% points of 420, 450, 495, and 570 F. Pertinent inspection tests for all special fuels are shown in Table I.



■ Fig. 8 – Effect of API gravity on exhaust smoke – Engine A – 1400 rpm full load

These fuels were subjected to the following testing schedule in Engine A devised as a result of previous experience:

(1) During 18 min idling at 350 rpm, economy and ex-



■ Fig. 7 – Effect of cetane number on exhaust smoke – Engine A – 350 rpm idle

haust smoke and odor were determined.

(2) Immediately following this idling period, maximum smoke was observed during full-throttle acceleration from idling to 1400 rpm.

(3) Economy, power, and exhaust smoke were determined when equilibrium had been reached at 1400 rpm full load.

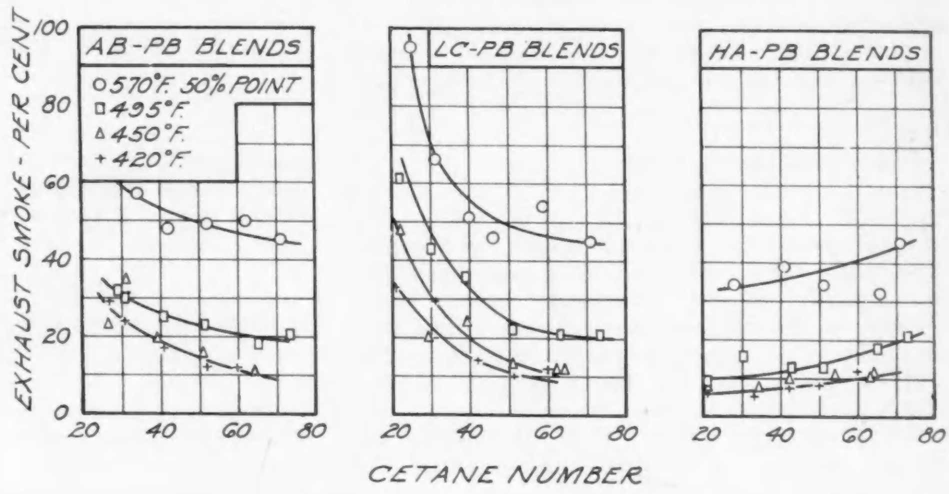
A summary of exhaust smoke results obtained on all special fuels is shown in Figs. 7 through 10. Both cetane number and volatility influence exhaust smoke at idling; an increase in either improves performance in this respect. This behavior is to be expected since idling smoke is of the white type previously shown to be affected by cetane number and volatility. Gravity appears to be the controlling factor with respect to full-load exhaust smoke, an increase in API gravity (decrease in density) resulting in reduced smoke. In this instance smoke is of the black type and as a result of insufficient oxygen to complete combustion. Since the air charge per cycle is substantially constant and since a lower mass of fuel is injected as the API gravity is increased, less smoke would be expected with this leaner fuel-air ratio. Apparently the carbon-hydrogen ratio does not change appreciably within each series of blends considered.

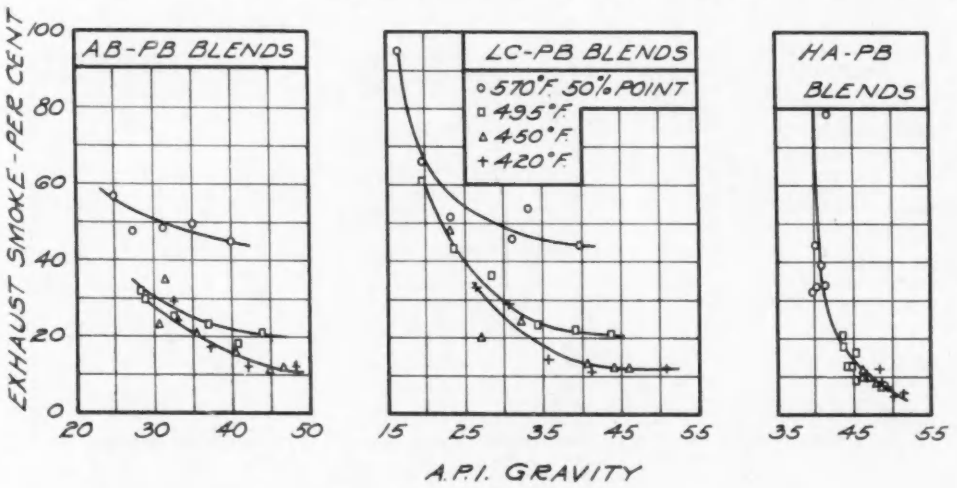
Figs. 9 and 10 show that gravity, volatility and cetane all influence maximum smoke during acceleration. The probable reason for this complex relationship is that this smoke is presumably composed of both types, black smoke due to rich mixtures at full load and white smoke from vaporization of liquid fuel in the exhaust system and poor combustion during the initial stages of the acceleration. It was previously shown that cetane and volatility affect smoke under low-output conditions and that gravity affects full-load smoke. With the exception of HA-PB blends, smoke is reduced by increased volatility, cetane, or API gravity. In the case of HA-PB blends, an increase in cetane results in an increase in smoke. This is undoubtedly due to the fact that with these fuels an increase in cetane number is accompanied by a decrease in API gravity, the latter apparently offsetting advantages derived through an increase in the former.

Cetane number was the only fuel property found to influence exhaust odor under the idling conditions employed; an increase in cetane improves performance in this respect. Fig. 11 shows typical results as measured by both aldehyde content and smell rating for all three fuel types having the same ASTM 50% point. A composite plot of all aldehyde data obtained is shown in Fig. 12. With the exception of very low cetane fuels where operation was invariably erratic, no differences, unexplainable by cetane number, are evident.

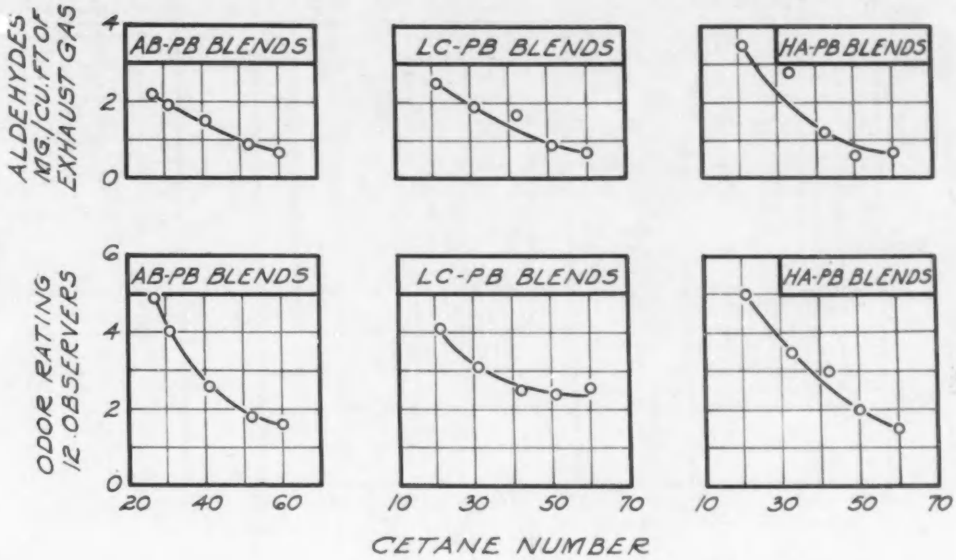
Gravity appears to be the controlling factor with respect

▪ Fig. 9 – Effect of cetane number on exhaust smoke – Engine A – full-throttle acceleration to 1400 rpm

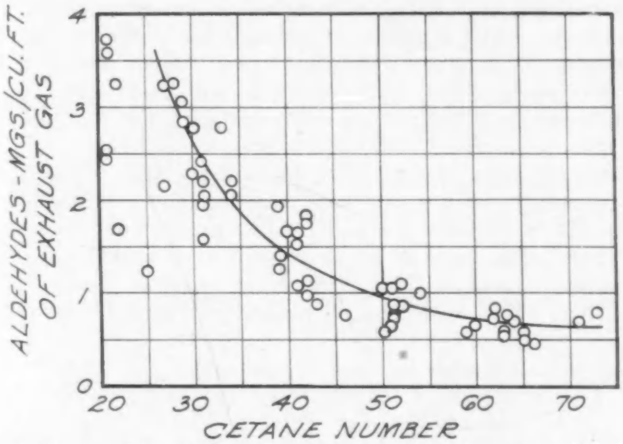




▪ Fig. 10 – Effect of API gravity on exhaust smoke – Engine A – full-throttle acceleration to 1400 rpm



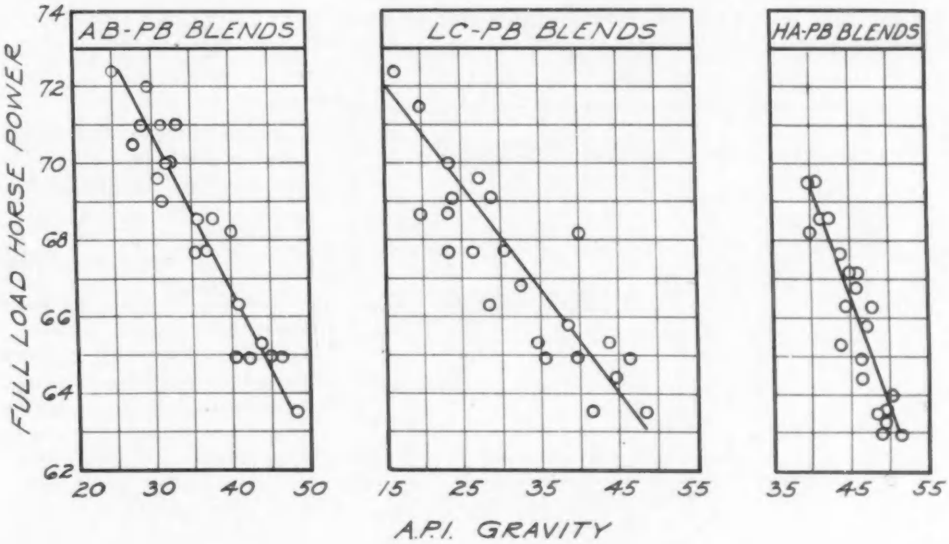
■ Fig. 11 – Effect of cetane number on exhaust odor – fuels of 420 F ASTM 50% point – Engine A – 350 rpm idle



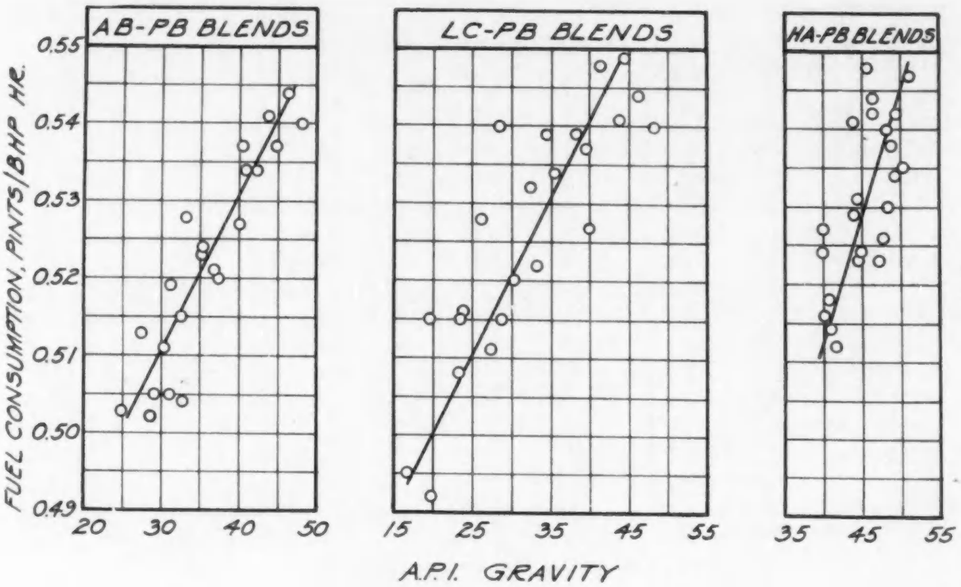
■ Fig. 12 – Effect of cetane number on aldehyde content of exhaust gas – Engine A – 350 rpm idle

to maximum power and economy as shown in Figs. 13 and 14. A decrease in API gravity results in increased power and economy, no doubt due to the increased volumetric heating values (more available energy per cycle) associated with heavy gravities. Fuel consumption at idling was found to depend only on cetane number as shown in Fig. 15. Since cetane number is a reflection of the ability of the fuel to support combustion under low-temperature conditions, high cetane fuels should require less fuel as observed.

In addition to the influence of API gravity on power and economy at full load, the effect of fuel viscosity was considered, since this will influence slippage past the injector plunger. Table 2 shows the effect of fuel viscosity on fuel injection rate for the special fuels. Although high viscosity appears to favor higher rates of fuel injection, the differences seem to be of minor importance, particularly for AB-PB and LC-PB blends. Worn injectors would un-



■ Fig. 13 – Effect of API gravity on full-load horsepower – Engine A – 1400 rpm



■ Fig. 14 – Effect of API gravity on fuel economy – Engine A – 1400 rpm full load

Table 2 – Effect of Viscosity on Quantity of Fuel Injected at 1400 RPM, Full Load, Manufacturer's Settings – Engine A

ASTM 50% Point, F	Range of Viscosities – S. U. Viscosities at 100 F	Average mm³ Fuel Cylinder Cycle
	AB-PB Blends	
420	31.2 - 33.2	66.2
450	32.2 - 35.0	66.5
495	35.0 - 37.0	67.2
570	42.3 - 51.6	67.9
	LC-PB Blends	
420	31.2 - 31.4	66.0
450	32.2 - 32.3	66.5
495	34.8 - 35.0	66.0
570	42.3 - 46.0	67.5
	HA-PB Blends	
420	31.2 - 33.5	64.3
450	32.2 - 35.8	64.6
495	35.0 - 41.2	65.0
570	42.3 - 69.4	67.5

doubtedly show much greater differences in injection rates than those observed in the present work.

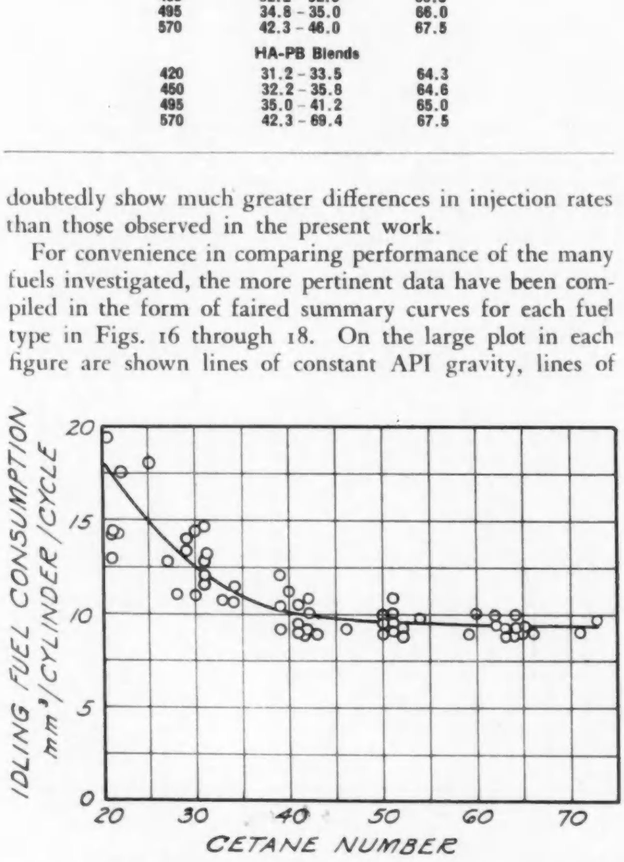
For convenience in comparing performance of the many fuels investigated, the more pertinent data have been compiled in the form of faired summary curves for each fuel type in Figs. 16 through 18. On the large plot in each figure are shown lines of constant API gravity, lines of constant maximum smoke accelerating and lines of constant aldehydes at idling. For the sake of simplicity, power, economy, and full-throttle smoke data are shown in side graphs. Table 3 is a comparison of the various phases of performance of the three fuel types, with respect to the fuel properties believed to be responisble for the performance observed; lowest numbers indicate best performance. On the basis of comparison, used HA-PB (paraffin) blends are, in general, superior, followed in order by AB-PB (naphthene), and LC-PB (unsaturate) blends.

Figs. 16 through 18 are of value in predicting the performance of other fuels of the same types as those examined. The AB-PB blends most nearly represent commercial fuels sold for high-speed diesels, and direct use of Fig. 16 might safely be made in most cases.

The use of Figs. 16 through 18 can best be illustrated by examples. Assume that it is desired to determine probable performance characteristics of three fuels showing the following inspection tests:

Fuel	Cetane No.	ASTM 50% Point, F	Gravity, deg API
X	45	525	33.0
Y	45	500	31.5
Z	45	550	42.2

It first must be determined which of the three charts applies in each case. Considering first the case of Fuel X, it is found that cetane number, ASTM 50% point, and API

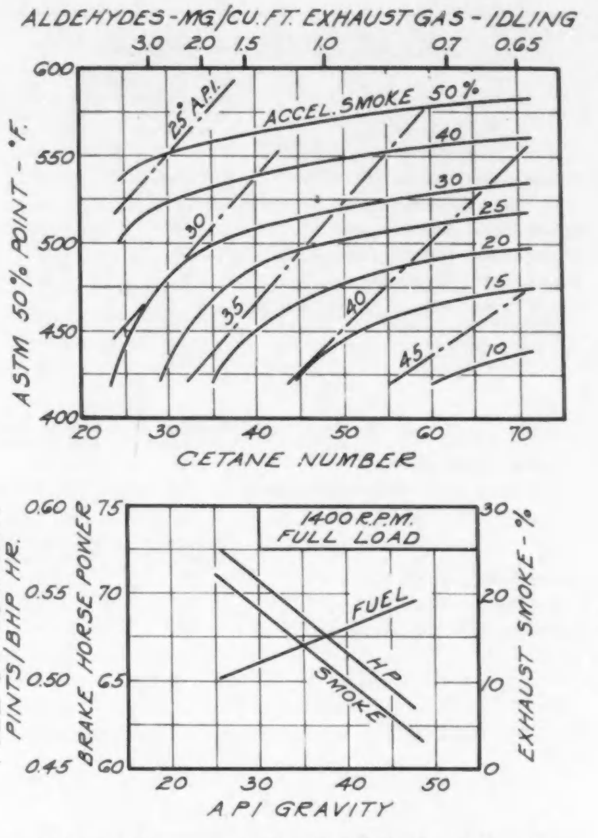


■ Fig. 15 – Effect of cetane number on idling fuel consumption – Engine A – 350 rpm

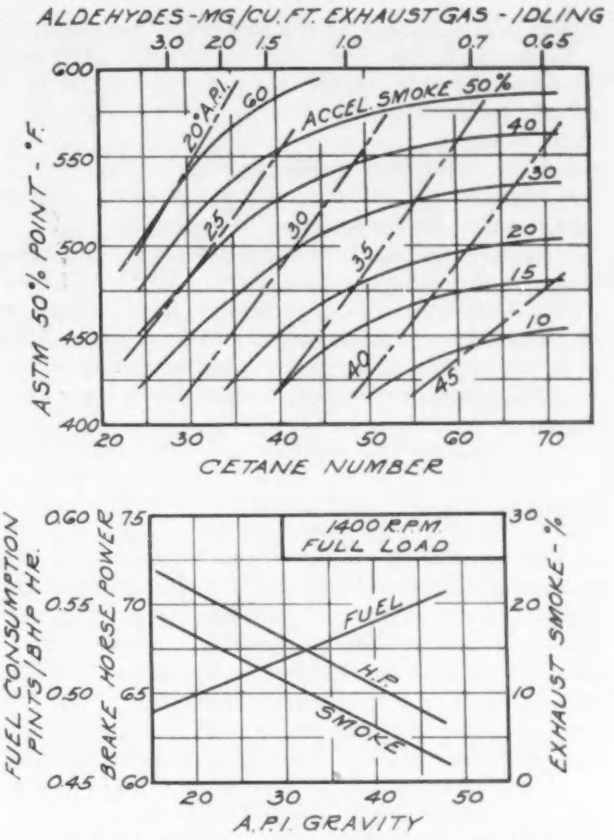
Table 3 – Relative Performance of Special Fuels *

Phase of Performance Considered	Fuel Property Basis Comparison	AB-PB Blends	LC-PB Blends	HA-PB Blends
Exhaust Smoke, Idling	Cetane, Volatility	2	3	1
Exhaust Smoke, Full Load	Gravity	2	1	3
Exhaust Smoke, Accelerating	Cetane, Volatility	2	3	1
Exhaust Smoke, Accelerating	Cetane, Gravity	1	1	—
Exhaust Smoke, Accelerating	Volatility, Gravity	1	1	—
Exhaust Odor	Cetane	1	1	1
Maximum Horsepower	Gravity	2	3	1
Full-Load Fuel Economy	Gravity	2	3	1
Idling Fuel Consumption	Cetane	1	1	1

* Lowest number indicates best performance.



■ Fig. 16 – Summary results, AB-PB blends – Engine A

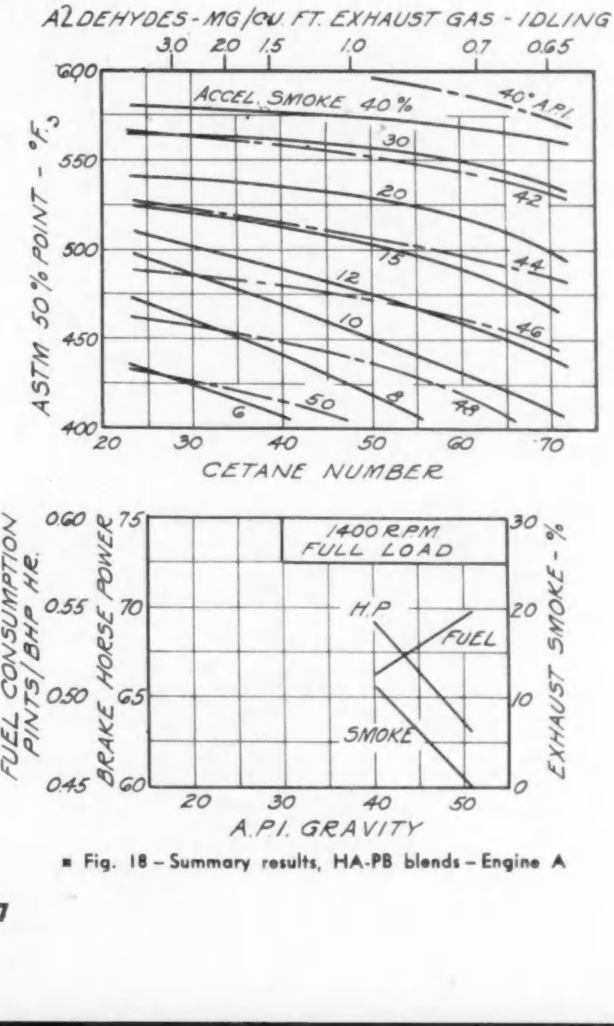


■ Fig. 17 – Summary results, LC-PB blends – Engine A

gravity can be plotted as a single point only in the upper chart of Fig. 16. This figure therefore applies to Fuel X. Similarly, it is found that Figs. 17 and 18 apply to Fuels Y and Z, respectively. Having established the applicable figures, exhaust-gas aldehydes and accelerating smoke are read directly from the upper chart, while full-load smoke, power and fuel economy are read from the lower at the appropriate API gravity value. Values so determined are tabulated in the following. If desired, idling exhaust smoke and fuel consumption can also be determined from the appropriate charts in Figs. 7 and 15, respectively.

	Fuel		
	X	Y	Z
Aldehydes, mg/ft^3	1.2	1.2	1.2
Exhaust Smoke, Accelerating, %	33	28	27
Exhaust Smoke Full Load, %	16	11	9
Full-Load Horsepower	69	68	68
Full-Load Economy, pt/bhp-hr	0.52	0.52	0.52
Exhaust Smoke, Idling, %	2	2	0
Idling Fuel Economy, mm^3/cyl/cycle	9.7	9.7	9.7

During the course of the work on Engine A, the question frequently arose as to the effects of fuel properties on engines embodying different design features. In order to check this point, a limited investigation was carried out on Engine B; testing was confined to the 22 AB-PB blends. The operating schedule was the same as that for Engine A except that, in order to establish equilibrium, a 25-min idling period at 450 rpm before acceleration rather than 18 min at 350 rpm, was used.



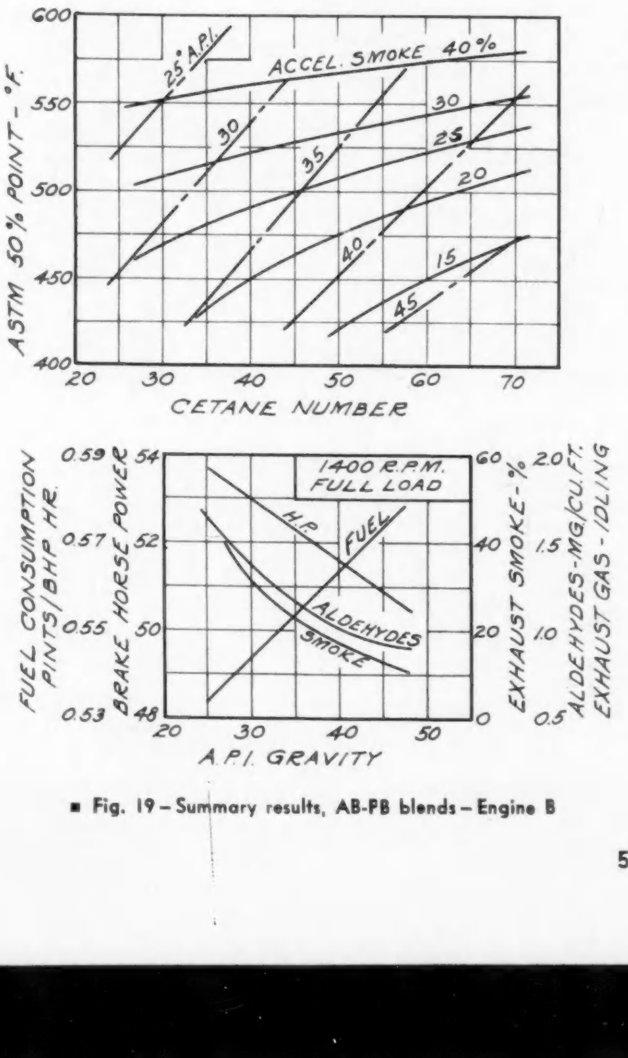
■ Fig. 18 – Summary results, HA-PB blends – Engine A

With few exceptions, results obtained are in qualitative agreement with those of Engine A. Summary data compiled from faired curves are shown in Fig. 19. Points of difference between fuel performance in the two engines are as follows:

1. None of the fuels smoked at idling in Engine B.
2. No difference between fuels was observed with respect to idling fuel consumption in Engine B.
3. Exhaust odor was of a much lower order in Engine B, even with fuels of very low cetane.
4. In Engine B. volatility as well as cetane number appeared to influence exhaust odor at idling. In view of this result, odor performance has, for convenience, been expressed in terms of API gravity since gravity bears some relationship to both of the influencing factors.

■ Engine Condition

It was pointed out previously that the engines used in the current work were maintained in first-class mechanical condition, and hence the aforementioned results can be expected only on equipment in similar condition. It appears obvious that poor maintenance practices such as improper injector and governor adjustments, worn or dirty injectors, or high oil consumption, will have a profound influence on engine performance, in many cases overshadowing any beneficial effects derived from premium fuels. In any case it is to be emphasized that good maintenance practices are necessary to take full advantage of fuels selected by operators to best meet their particular requirements. Table 4 will serve to demonstrate the effect of mechanical condition on engine performance.



■ Fig. 19 – Summary results, AB-PB blends – Engine B

Table 4 – Effect of Engine Condition on Performance

Performance Feature	Engine A: Injectors after Extensive Use in Bus Service	Engine A: Same Injectors after Overhaul
Exhaust Smoke – 350 rpm Idle, %	2	0
Exhaust Smoke – 1400 rpm No Load, %	2	0
Exhaust Smoke – 1400 rpm 1/4 Load, %	8	6
Exhaust Smoke – 1400 rpm 1/2 Load, %	11	8
Exhaust Smoke – 1400 rpm 3/4 Load, %	16	14
Exhaust Smoke – 1400 rpm Full Load, %	41	37
Exhaust Smoke – 1400 rpm 3/4 Load, after 15 min at 350 rpm Idle, %	37	18
Fuel Consumption at 1400 rpm No Load – mm^3/cyl/cycle	21.2	18.0
	Engine A: Governor in Bad Repair	**Engine A: Same Governor after Overhaul**
Exhaust Smoke Decelerating, %	70	0
Aldehydes – mg/ft^3 Exhaust Gas Decelerating	0.9	0
	Engine B: Engine after Extensive Use	**Engine B: Same Engine after Major Overhaul**
Exhaust Smoke – 1400 rpm Full Load, %	34	20
Exhaust Smoke – 2500 rpm Full Load, %	44	16
Exhaust Smoke – Accelerating, %	38	16

■ Conclusions

It appears that fuels may be chosen to give desired results along any selected lines of engine performance, but it is not possible to obtain all desirable results with a single fuel, at least not without recourse to additives. For example, an increase in cetane number or volatility will reduce exhaust smoke, and an increase in cetane number will reduce exhaust odor. However, an increase in either of these fuel properties is accompanied by decreased power and economy. The only present solution appears to be fuel selection on the basis of a compromise amongst the various factors involved. Proper interpretation of the engine performance characteristics most desired and consideration to the type and mechanical condition of the engine used should result in selection of fuels for utmost satisfaction. No doubt future engine design will obviate the necessity for tailor-made fuels and precise maintenance practices.

■ Acknowledgment

The authors acknowledge their appreciation to C. E. Cummings and E. M. Barber for helpful suggestions; to W. S. Bailey and W. J. Kenney and their assistants who did a large part of the experimental work.

Appendix I

EXHAUST-GAS SAMPLING AND ANALYSIS

(Mikita, Levin, Kichline)[3]

Sampling Exhaust Gas – Samples for odor rating and aldehyde determination were obtained with the equipment shown in Fig. 20. Gas for aldehyde determination was taken from the exhaust pipe a short distance from the end of the manifold *A*, passed through a water-cooled condenser *B*, three traps immersed in dry ice and chloroform *E*, and through an orifice-type meter *F*. The gas was

drawn at the rate of approximately 0.5 cfm for fifteen 1-min intervals, the condensate being ample for the test. Preliminary experiments demonstrated that the portion of exhaust gas which passes uncondensed through this train does not have the disagreeable odor. The sample taken for chemical determination of aldehyde was the combined condensates from the condenser and traps, being blended after rising to approximately room temperature.

Samples for odor intensity rating by smell were taken from the exhaust pipe by passing the gas through a quart bottle H immersed in cracked ice and water G, and through an orifice meter F_1, the gas and condensate remaining in the bottle being used.

Determining Relative Odor Intensity by Smell – The quart samples were rated in the following manner: Four or five of them were given to an individual to be rated for intensity as 1, 2, 3, and so on, 5 being the most intense and 1 the least; these samples were then given to four other individuals for rating; the accepted nasal rating was the average for the five individuals.

It is interesting that a particular technique was necessary to obtain consistent intensity ratings. If the tester first inhaled deeply of a very strong sample, the membranes of the nose became so irritated that further ratings were, for some time, impossible. The best procedure was to uncork a bottle, bring the nose to approximately 1 in. from the mouth of the bottle and inhale gently. Before smelling the next sample a pause of approximately 5 sec was advisable. In a group of four samples the most and least intense were readily rated; the intermediate two require more attention.

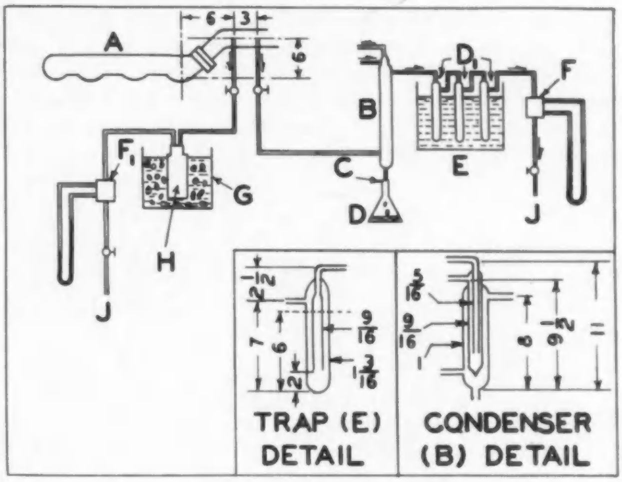
Chemical Test for Exhaust Odor – In the course of the engine tests to determine the conditions of operation affecting odor of exhaust gas, it was soon found impractical to depend on one's nose for rating the necessary number of samples and an impersonal method was sought. Preliminary analyses showed that, when a gasoline bus operates under conditions of deceleration, the pungent and disagreeable exhaust always contains aldehydes in appreciable quantity. Other substances are also found, such as lower fatty acids, oxides of nitrogen, incompletely burned fuel, and so on, but it was evident that these substances did not impart the pungent odor being investigated. Chemical analysis showed that formaldehyde was present in every sample examined; acetaldehyde rarely; acrolein was never found; and a few other aldehydes which were specifically sought were either absent or the test obtained was so questionable as to be of no value. An attempt was made, therefore, to correlate aldehyde content, in terms of formaldehyde, with intensity of pungent odor in exhaust gas. For convenience in handling, the exhaust gas was condensed and this liquid used in the subsequent examinations. The condensates do not have as disagreeable an odor as uncondensed exhaust gas; however, nasal ratings of samples in a group regularly placed them in the same relative order whether they were collected as gas or condensed.

Determining Aldehydes in Exhaust Gas – It was convenient to employ the reaction with Schiff's[4, 5] reagent for the aldehyde determinations. The intensity of the reaction color, which is related to aldehyde content, was determined by means of Yoe[6] photo-electric colorimeter.

Initial tests were made by measuring the time required for samples to develop a predetermined color intensity, but this method proved unreliable and not flexible enough to cover the range of samples involved. Further experiments with a series of solutions of known formaldehyde content showed good correlation between color intensity and aldehyde content, provided a fixed time and constant temperature were used in developing the reagent-aldehyde reaction product. The actual formaldehyde content of these solutions was derived from chemical determinations on a concentrate by the method of Ripper[7], the concentrate being subsequently diluted with known amounts of water. A curve was drawn plotting micro-amperes shown by the Yoe instrument versus formaldehyde content and, from this curve, the values of unknowns were read. Details of the procedure for determining aldehydes follow.

Reagents – Modified Schiff's Reagent[8] – Dissolve 0.2 g of Kahlbaum's rosaniline hydrochloride in 120 cc of hot water. Cool and add 2 g of anhydrous sodium sulfite dissolved in 20 cc of water, followed by 2 cc of concentrated hydrochloric acid. Dilute to 200 cc and store in glass-stoppered amber bottle.

Apparatus –
Constant-temperature bath at 100 F
Test tubes, ¾ by 6 in.
Photo-electric colorimeter, (Yoe)



■ Fig. 20 – Collection of exhaust sample (Dimensions in inches)

A – Exhaust manifold
B – Condenser (water cooled)
C – Gas-tight connection
D&D₁ – Condensate sample
E – Traps in bath of dry ice & chloroform (—60 F)
F&F₁ – Orifice meters
G – Bath of cracked ice and water
H – Sample for smell
J – To vacuum line

[4] See p. 1475 "Standard Methods of Chemical Analysis," by W. Scott, Fourth Edition, 1925, D. Van Nostrand Co., New York.
[5] See the *Journal of the American Chemical Society*, Vol. 30, 1908, pp. 1607-1611: "The Colorimetric Estimation of Benzaldehyde in Almond Extracts," by A. G. Woodman and E. F. Lyford.
[6] See *Industrial and Engineering Chemistry*, Analytical Edition, Vol. 7, July 15, 1935, pp. 281-284: "A Photoelectric Colorimeter," by John H. Yoe and Thomas B. Crumpler.
[7] See p. 374, "Volumetric Analysis," by F. Sutton, Eleventh Edition, 1924, R. Blakiston's Sons & Co., Philadelphia.
[8] See *Industrial and Engineering Chemistry*, Vol. 18, 1926, pp. 304-306: "Detection of Methanol in Alcoholic Beverages," by F. R. Georgia and Rita Moralis.

Table 5 – Exhaust Smoke and Odor and Fuel Consumption – Commercial-Type Fuels – Engine A

Fuel No.	12	4	7	2	5	11	1	10	3	8	6	9	13
Cetane No.	72	57	56	50	48	48	47	44	44	40	38	33	21
ASTM 50% Point, F	493	477	523	417	416	547	502	412	504	600	621	500	480
API Gravity, deg	44.0	43.7	37.5	43.0	43.3	32.5	35.4	40.8	35.2	26.7	23.1	29.0	21.3
Per Cent Exhaust Smoke – Equilibrium													
350 rpm idle	0	0	1	0	0	1	0	0	0	4	25	10	68
1400 rpm no load	0	0	0	0	0	0	0	0	0	5	25	8	70
1400 rpm full load	35	—	24	16	—	45	20	8	—	26	37	27	36
Per Cent Exhaust Smoke – Dynamic													
Max. smoke at 1400 rpm full load after 15 min misfiring[1] at 1400 rpm	33[3]	99[2]	99[2]	35	37	97[2]	90[2]	24	94[2]	97[2]	99[2]	97[2]	—
Max. smoke at 1400 rpm ¾ load after 25 min idling at 350 rpm	31	18	35	8	6	26	10	11	12	45	58	20	70
Aldehydes – mg/ft³ of exhaust gas													
350 rpm idle	—	0.7	1.2	0.6	1.1	0.8	1.0	—	1.7	1.0	2.1	1.8	3.9
1400 rpm no load	0.7	1.3	1.5	2.4	2.1	1.6	2.2	2.5	3.5	2.9	3.3	4.5	4.8
No-Load Fuel Consumption – mm³/cyl/cycle at 1400 rpm	15.5	17.0	17.0	18.2	17.5	17.5	18.0	21.6	19.5	20.5	22.3	26.0	42.0

[1]—Fuel injection rate of 11.7 mm³/cyl/cycle.
[2]—Large quantities of liquid fuel present in exhaust system.
[3]—No misfiring occurred with this fuel.

Pipettes, 2 ml
Pipette, 10 ml, graduated in 0.1 ml

Procedure – Two ml of exhaust-gas condensate is introduced into a test tube and 8 ml of Schiff's reagent added. The test tube is stoppered and placed in a bath at 100 F for 15 min to develop the reaction color. The mixture is then poured into the Yoe colorimeter tube, the test tube rinsed with 5 ml of water at 100 F and this added to the colorimeter tube, the solution being further mixed by gentle shaking. The colorimeter tube and its contents are immediately placed in the photo-electric colorimeter and tested against a blank consisting of 13 ml of distilled water and 2 ml of sample. This type of blank was chosen because exhaust-gas condensates are frequently cloudy and it is best to duplicate this cloud in the blank. Employing the colorimeter in the manner described by Yoe and Crumpler,[6] micro-ampere readings are taken and expressed in terms of formaldehyde by reference to the calibration curve. In the standardization of our instrument the following table of values was obtained:

% Formaldehyde (in solution under test)	Micro-amp
0.0010	48.3
0.0019	47.6
0.0031	46.2
0.0097	39.8
0.0139	35.5
0.0323	25.8
0.0485	22.7
0.097	17.8
0.323	13.5
0.97	11.3

The data were plotted on semi-log paper to give a suitable slope to the calibration curve and better spacing in the lower concentrations where most samples fall.

Since Schiff reagent is rather unstable, it should be checked periodically (preferably daily) against solutions of known formaldehyde content. In duplicate tests on the same condensates, results were reproduced within two in the fourth decimal place on samples of low concentration.

It was found that, if a number of individuals smelled a series of four bottles of exhaust gas obtained from engines run under controlled conditions producing an assortment of intensities of pungent odor, all rated the extreme samples consistently but the order of the intermediate samples was frequently reversed. The aldehyde tests always checked the nasal ratings on extreme samples and duplicate aldehyde tests rated intermediate samples in consistent order. This is significant since it demonstrates that other aldehydes which may be present do not alter the correlation between odor and the chemical test based on formaldehyde. Similar correlation between smell and chemical test was obtained on solutions of known strength prepared from pure formaldehyde.

Appendix II

ADDITIONAL PERFORMANCE DATA – COMMERCIAL FUELS

Table 5 is a summary of additional exhaust smoke and odor and fuel consumption data obtained on commercial types of fuel in Engine A, under both equilibrium and dynamic operation.

It is indicated that, under no-load conditions, low cetane and low volatility fuels favor smoky exhaust. In the case of full-load operation, heavy gravity fuels appear to produce more smoke than those of lighter gravity. Under dynamic conditions simulating acceleration after low output, both volatility and cetane affect exhaust smoke. In the case of the very high-cetane fuel, No. 12, misfiring did not occur; hence little if any liquid fuel was collected in the exhaust system. In the case of the very volatile fuels Nos. 2, 5, and 10, though misfiring occurred, little fuel was collected in the exhaust system. In these cases a low order of exhaust smoke was observed.

With no-load operation both aldehydes in the exhaust gas and fuel consumption are apparently dependent only upon cetane number.

DYNAMICALLY STABLE SPRING SUSPENSION for Railway Cars and Motorcoaches

by PAUL K. BEEMER[1] and F. C. LINDVALL[2]

ANY land vehicle necessarily has its center of gravity above the point of support on the road or rail surface. The height of the center of gravity above the road surface in relation to the wheel tread fixes the maximum overturning side force in a simple geometrical way which is independent of the internal arrangement of axle, springs, hangers or other parts comprising the vehicle mounting on its wheels. The dynamical behavior of the vehicle, however, the riding qualities, and the general roadability are controlled definitely by the spring and body support arrangement.

Railroad equipment, unlike automotive vehicles, is rigidly held to standard track gage and, because of requirements for interchangeability, is almost as rigidly restricted as to height of center of gravity above the rail. Moreover, side clearances for rolling stock prevent much departure from standard truck dimensions. As a result, new concepts of truck construction, body suspension, and springing must be within the limiting dimensions which are now standard. Highway motor bus construction too, is limited in maximum width, but wheel tread is variable and full clearance width may be carried nearly to the road surface, apparently permitting a wider spacing of springs and greater lateral stability than would be possible within the dimensions of a standard railway truck. However, much of the bus width at the axle is taken by the dual tire arrangement in general use with the result that the springs are much

THIS paper describes an "above-gravity, dynamically stable" spring suspension in which the car or coach is elastically supported at each end on a virtual, universal center bearing on the longitudinal centerline above the center of gravity of the car or coach body. It is pointed out that this virtual support permits sufficient universal swivel action of the truck relative to the car body to account for all operating conditions, and the springs, together with the positioning linkage, cooperate to achieve the desired vibration isolation. The actual car support, however, is at two points on either side of the car centerline with a third attachment below the floor level.

The car or coach body rests on soft coil springs which are recessed into the car structure on either side of the center aisle. These springs carry only vertical load and allow, within limits of safe stress, free horizontal movement of the top relative to the bottom for all lateral and turning movements of the truck in normal service. Lateral movement of the body floating on the main springs is restrained by control arms and links which act on the body above the center of gravity. The longitudinal position of the truck is maintained by the thrust tube or "wagon tongue" which is anchored in rubber at one end of the truck frame and at the other to the car underframe.

■ ■ ■

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[This paper was prepared for presentation at the 1942 Semi-Annual Meeting of the Society, which was ruled out because of transportation priorities.]

closer together than desirable for the elimination of body roll. Further discussion of body roll is given in the general considerations of riding and stability which follows.

Any vehicle traveling over an imperfect road surface must have its body supported by elastic members if the irregularities encountered are not to be transmitted in their full severity to the body structure and its occupants. The support of the body on the wheels must provide not only adequate isolation against these vertical and lateral wheel motions, but also stability under all ordinary operating conditions. As is well known, the body may be isolated against vertical vibrations by using springs having a large static deflection. A 10-in. deflection is common for railway passenger cars and gives a resonant frequency of 1 cps. The transmissibility of such a system, that is the ratio of the body motion to the wheel motion, is low for all track irregularities which occur at high speed, and a high degree of comfort results.

The wheels of a railroad passenger truck, in rolling along a track, are constrained to follow all the vertical and lateral irregularities existent in the tracks. These irregularities may be minimized by careful and expensive maintenance of way, but never eliminated completely. Additional lateral wheel motion occurs as oscillation within the limits of flange clearance, induced by wheel-tracking errors and truck nosing. Inadequate curve compensation and improper rail alignment on curves also give rise to lateral forces which seriously affect the stability of the car body mounted on trucks in the conventional manner.

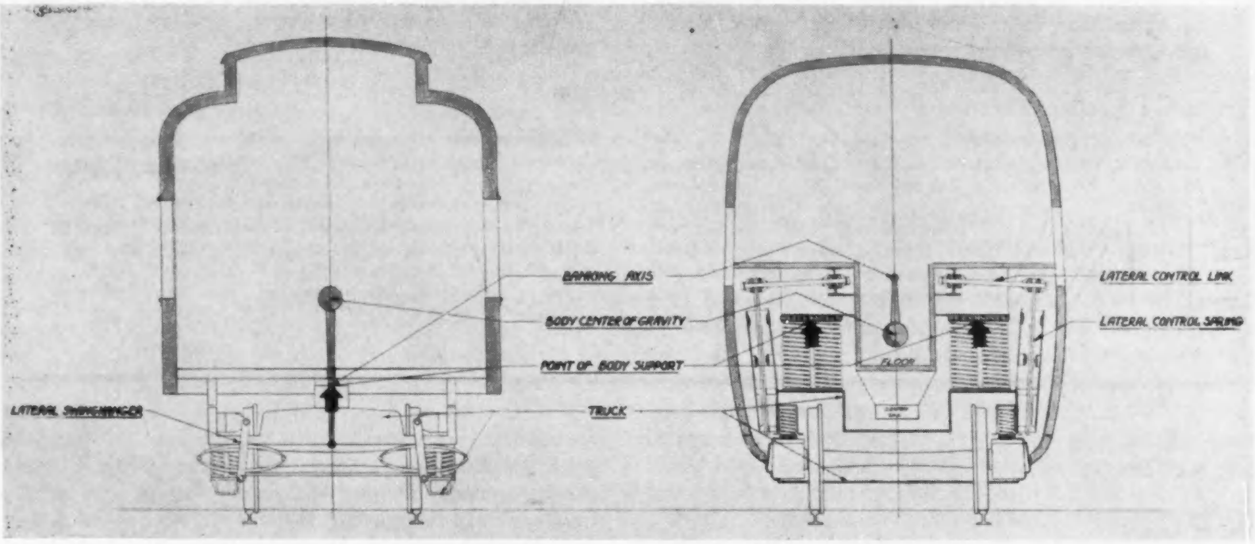
In conventional railroad-truck and motor-vehicle constructions, the springs are at a considerable distance below the center of gravity of the body, and instability in the form of objectionable and even dangerous body roll results from the use of "soft" springs. See Fig. 1 (left). Precisely this experience in the automotive industry followed the use of springs of greater static deflection. Body roll, particularly on curves, became objectionable, and has been minimized by cross-stabilizers or "sway bars" designed to give the body freedom of vertical motion but restricted roll. This scheme has been incorporated in some recent railroad trucks in an effort to control the instability of soft truck springs. However, some of the benefit of soft springs is lost because, on standard track with staggered rail joints, free truck roll is desirable and the stabilizers act to stiffen the springs for this type of action.

Automotive experience, particularly with passenger cars, also has shown that the addition of stabilizers or "sway bars" has the undesirable effect of increasing "boulevard jiggle"; therefore, some of the desirable effect of the soft springs now in use on the passenger cars is lost by the addition of the necessary stabilizers. A suspension which is inherently stable would obviously permit the use of soft springs without the necessity of adding such restricting devices.

Lateral freedom of a railroad truck suspension system is also desirable in order that the wheels may be free to follow the track irregularities without transmitting these motions to the car body. Automotive equipment need not have this flexibility, because the wheels are not obliged to follow in a fixed track on the highway; consequently the only lateral forces which are introduced are those of accidental surface irregularities which can be accommodated largely by tire deflection. In the standard railway truck the swing hangers provide lateral freedom but, due to space limitations, are rather inflexible in design.

In view of the limitations of the conventional railroad truck, the object of this development has been to produce a car-body suspension system which would provide the requisite isolation against vibration and maintain stability. The goal has been comfort at high speed on ordinary track with safety and economy of weight. The fundamental scheme has been to support the car above its center of gravity in a manner which is inherently stable. Ideally, the above-gravity suspension hangs the car above its center of gravity on an imaginary longitudinal axis which is allowed all necessary vertical and lateral movement against soft spring restraints. The wheels follow the rail irregularities while the body floats about a central position.

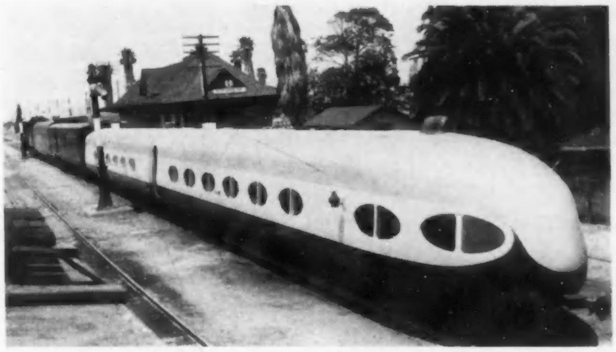
The above-gravity suspension has a further outstanding advantage over the conventional truck. In taking curves above super-elevation speeds, the outward force acting on the center of gravity causes the body of a conventional car



■ Fig. 1 – Diagrammatic comparison of conventional and pendulum trucks

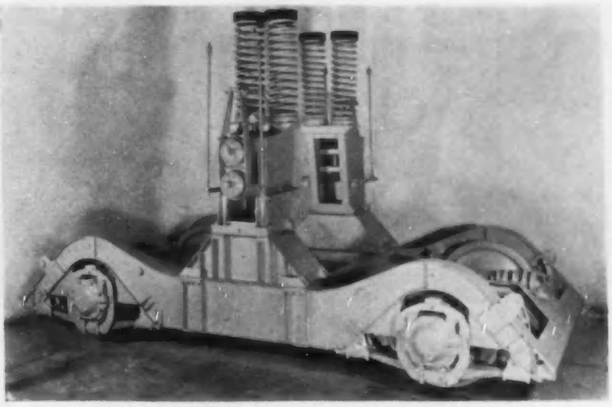
to roll outward on the truck springs, adding to the discomfort of the passengers. However, with the car supported above its center of gravity, the outward force rolls the body in the direction for comfort, pendulum-wise, adding effectively to the super-elevation of the track in so far as comfort and stability within the car are concerned.

A practical realization of the benefits of the above-gravity suspension was the purpose of the design and experimental work leading to the construction of the experimental cars shown in Fig. 2. The original designs and the subsequent modifications adhere to a basic, ideal car mounting in which the car is elastically supported at each end on a virtual, universal center bearing on the longitudinal centerline above the center of gravity of the car body. This virtual support permits sufficient universal swivel action of the truck relative to the car body to account for all operating conditions, and the springs, together with the positioning linkage, cooperate to achieve the desired vibration isolation. The actual car support, however, is at two points on either side of the car centerline with a third attachment between the truck and the car body below the floor level. Consequently, no objectionable interference with normal use of the car interior is introduced by the suspension. The desired motions are provided by flexure of the support system, suitably positioned and restrained. All motion between the truck and car body occurs solely through elastic flexure, leading to an extremely simple light-weight truck and suspension system, Fig. 1 (right).



■ Fig. 2 – Plywood experimental cars in test train

The rear truck of the experimental unit illustrates the basic principles involved in the suspension system and is shown in Fig. 3. The car body rests on soft coil springs which are recessed into the car structure on either side of the center aisle. These springs carry only vertical load and allow, within the limits of safe stress, free horizontal movement of the top relative to the bottom for all lateral and turning movements of the truck in normal service. A basic analysis of the mechanics of helical springs disclosed two facts which made the application feasible with properly designed springs; that is, the stresses, due to these lateral deflections, are not severe, and the springs can be deflected laterally and yet be stable. Lateral movement of the car body floating on the main springs is restrained by control arms and links which act on the body above the center of gravity. The control arms mounted in rubber give a variable spring rate for lateral motion, so that the car floats about a center position with small restraint, equivalent to the action of very long swing hangers, and is brought to a yielding stop for large lateral swings. The longitudinal position of the truck is maintained by the thrust tube or "wagon tongue" which is anchored in rubber at one end to the truck frame and at the other to the car underframe. The rubber mountings of this longitudinal tie permit lateral movement of the truck and angularity on curves as well as constituting barriers against noise transmission. The coil-spring suspension involves no sliding or rotating parts carrying the weight of the car. The elements which have been described replace the center plate, side bearings, bolster, chafing plates, bolster springs, spring plank, and swing hangers used in all standard passenger car trucks.

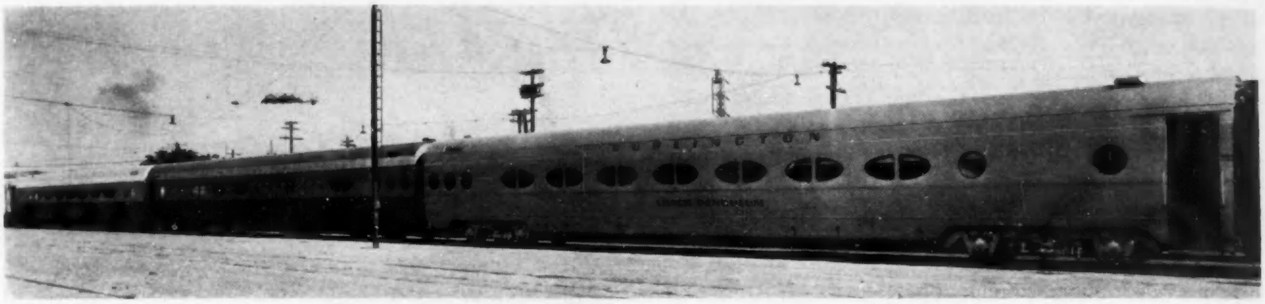


■ Fig. 3 – End truck used on experimental cars

The car bodies of the test unit were, for reasons of simplicity, fabricated of plywood into stressed-skin or semi-monocoque structures. High strength and rigidity were thus obtained at minimum weight. Moreover, most of the usual auxiliary equipment of passenger cars was omitted from these units so that the two-car articulated unit 145 ft in length weighed only 33 tons on the rail.

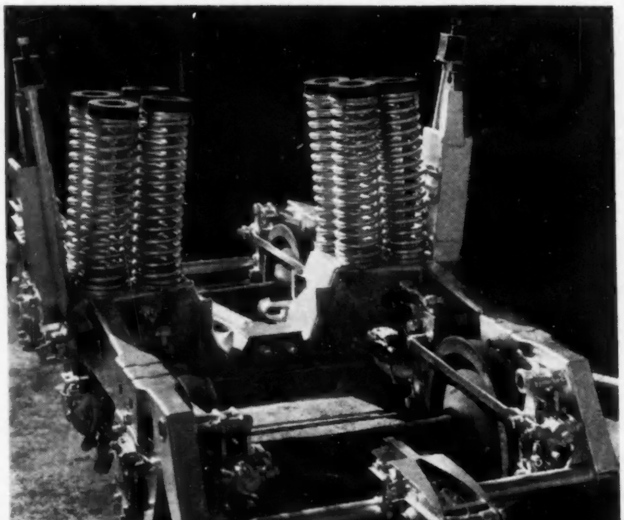
Despite their extreme light weight, these cars in road tests, at speeds up to 100 mph, demonstrated outstandingly better riding qualities than older, heavyweight equipment and modern, lightweight coaches utilizing conventional trucks. The demonstrations have verified conclusively the prediction of theory – that heavy equipment is not necessary for good riding. Static deflection of the spring system, whatever the car weight may be, determines the transmissibility of the system and, ultimately, the comfort of the passengers within.

Based upon the success of the experimental train unit, the Santa Fe, the Great Northern, and the Burlington roads each contracted for a deluxe-type coach incorporating the above-gravity suspension system and skin-stressed lightweight body structure (Figs. 4 and 5). The specifications required that the cars be built for unrestricted interchange service; hence one car with the new suspension system, when coupled in with standard equipment, is subject to end reactions which alter the performance of the spring system to a small degree. The weight of the service car is much greater than that of the experimental car, and the floor heights are different. Dynamically, however, the two types of car are similar. The relationship between the point of the lateral restraint and the height of the body center of gravity is the same in the new car as in the



■ Fig. 4 – Steel cars built for Great Northern, Santa Fe, and Burlington railroads

■ Fig. 5 – Interior of one of the new cars



■ Fig. 6 – Truck for steel car

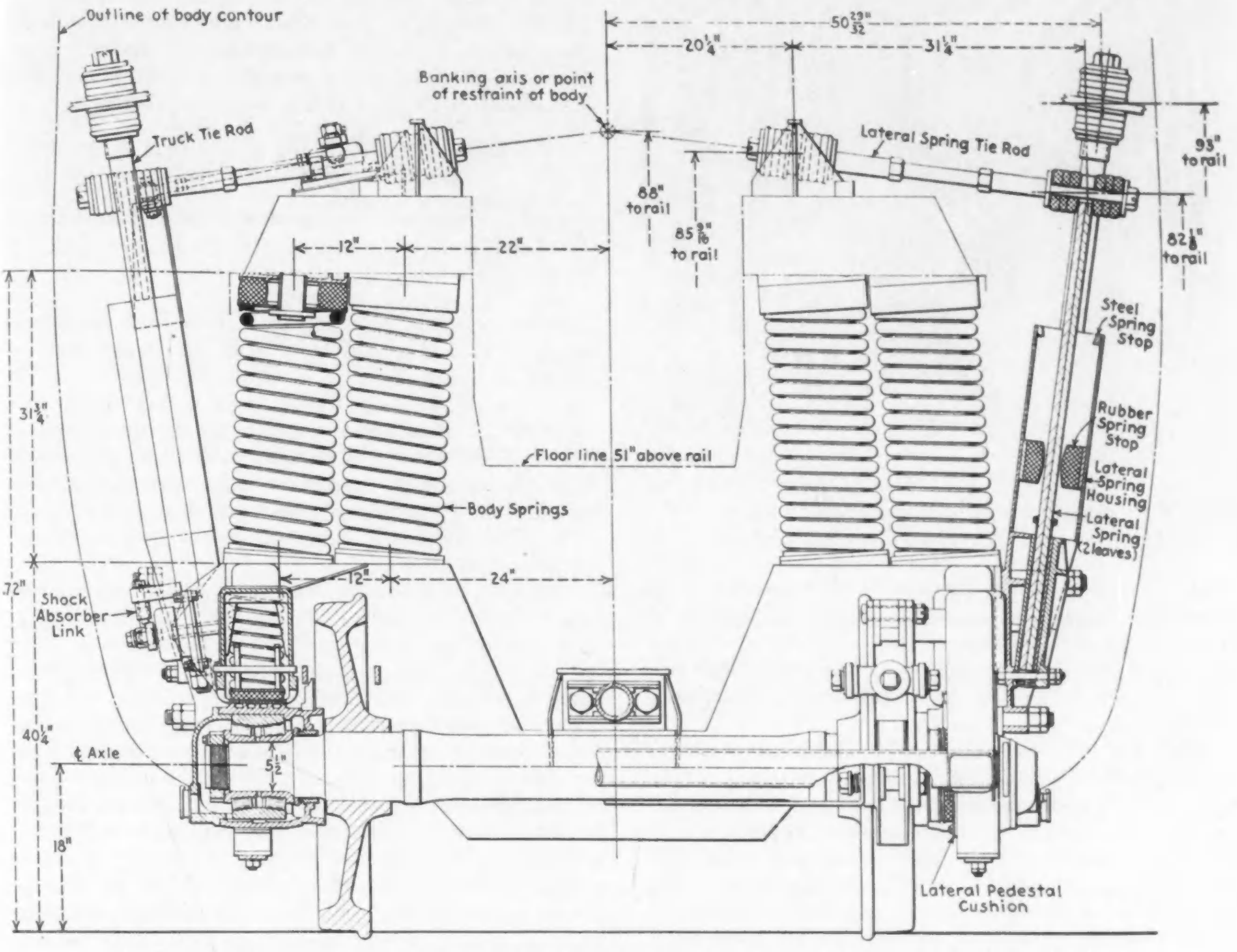
experimental unit. Similarly, the spring deflections and lateral spring rate found satisfactory in the test car are used in the new coaches. The successful performance of the experimental trucks indicated that the journal springs should be relatively stiff. The journal coil springs are mounted just above the boxes, and are applied so that some lateral movement can take place between the journal boxes and the truck frame. This movement is permitted by rubber and steel vulcanized pads on the side of the pedestals which are deflected in compression to relieve lateral shocks. The arrangement of parts is different from that used on the experimental cars but the characteristics are the same. The truck frames are arc-welded of high-tensile, low-alloy steel and are stress-relieved before machining. See Fig. 6.

Eight body springs are used per truck, four on each side. These springs are mounted just above the frame side members and extend upward 26 in. within the body to the body-support structure. The static deflection of 10 in., together with the rubber insulator at the top of the springs with a deflection of ⅜ in., insulates the car body thoroughly from disturbances in the truck. Considerable analytical and test work was performed to establish correct relationships between static deflection, working height, and pitch diameter of the springs to obtain freedom of lateral movement and stability. In these springs, the greatest lateral movement encountered in normal service increases the working stress near the ends of the coils by 25%.

The lateral springs consist of two plates clamped rigidly to the side of the truck frame, extending upward between the body-support bulkheads to a point above the center of gravity of the body. See Fig. 7. Rubber-cushioned progressive lateral stops shorten the effective spring length with increased deflection, and thereby increases the spring rate with deflection, as shown in Fig. 8. Lateral tie-rods with rubber-mounted end connections attach the lateral spring to the car body at a point about 20 in. above the center of gravity of the entire body assembly.

Hydraulic shock absorbers, mounted on the side of the truck frame, are connected by means of long vertical tie-rods to the body structure. These two vertical tie-rods are also used to hold the truck to the car body in case of derailment or overturning. Hydraulic shock absorbers, mounted in the body-support structure, are connected by means of lateral links to the tops of the lateral spring housings which are rigidly attached to the truck frames at the lower ends.

The truck thrust tube, used to position the truck longitudinally with respect to the body, is connected to the

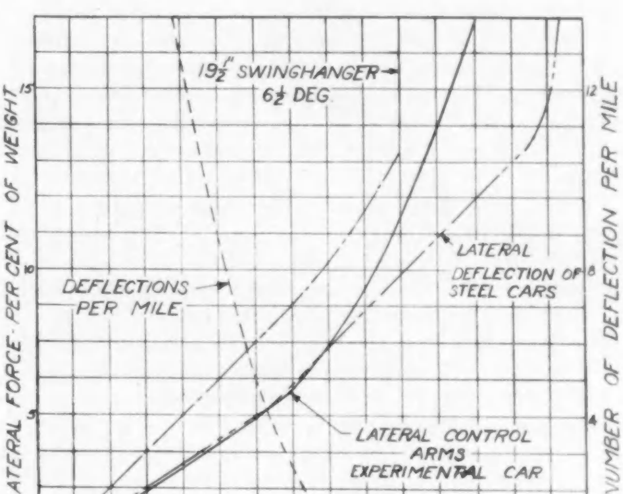


■ Fig. 7 – Partial cross-section showing truck details (Illustration furnished by "Railway Mechanical Engineer")

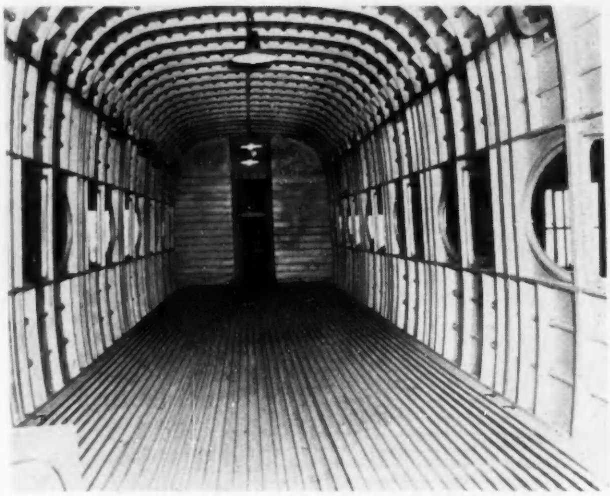
truck through a large rubber fitting at the center of the truck frame transom. The body connection of the thrust tube is made through a similar fitting attaching to a bracket mounted directly under the draft-gear pocket.

The essential structural elements of the steel stressed-skin construction embodied in the body of this car (Fig. 9) are: (1) thin skin or sheathing; (2) longitudinal stiffening members; (3) transverse stiffening carlines at the roof; side posts at the sides; and floor beams, or cross-bearers on the underframe; (4) structural bulkheads through which vertical and lateral loads are transferred to and from the body structure; and (5) miscellaneous structures, such as heavy end frames, essential for housing draft and buff gear; center sill, essential for heavy buff and draft loads; and the body spring-supporting elements.

The longitudinal and the transverse stiffening members are rigidly fastened to, and act integrally with, the skin. The longitudinally corrugated flooring is shear-connected into the structure, making the car body a huge tube, closed at both ends, with a high degree of torsional rigidity. The bending rigidity of the structure is likewise high since all the effective material is disposed as far from the



■ Fig. 8 – Characteristics of lateral springs



■ Fig. 9 – Steel body structure

neutral axis as possible. Elliptical windows reduce stress concentrations and increase the shear rigidity of the body.

The entire car body structure, with the exception of the arc-welded end framing, draft gear attachments and miscellaneous connections, is fabricated with controlled spot-welding.

Much attention was paid in the construction of these cars to acoustical treatment. An absorptive material was applied to the underside of the car body and to the recesses in which the springs mount to minimize the intensity of sound at its source. Furthermore, the basic suspension system of the car body permits a liberal use of rubber at all points of attachment between the truck and the car body, which eliminates the direct transmission of objectionable truck noises to the car body. As a result, these three new cars are noticeably more quiet than the conventional passenger cars of new design. This reduction of noise, together with the absence of annoying vibration, provides a new high in comfort.

An extensive road-test program with the experimental cars, during which various changes were made, showed that the only modifications desirable were reduction of shock-absorber control and application of softer rubber discs in the lateral-control arms to provide a lower lateral frequency.

During test runs, readings were taken by means of scratch gages to determine deflections of all rubber parts and springs. From previously obtained load deflection curves of each of these units, maximum loads were determined for all principal elements of the truck. Some of the runs were made at high speeds on a very rough secondary freight road. Cathode-ray-oscilloscope observations were made of the high-frequency vibrations throughout the truck suspension system at a variety of speeds. A very high value is now placed on the use of the rubber in some parts of the truck, as a result of these studies.

To record lateral shocks transmitted to the car body, a very simple and crude accelerometer was used. Small hardwood blocks, "tumble blocks," were carefully calibrated for tipover value on an inclined plane and were matched in two sets of three blocks each. During most of the test runs, one set of these blocks was operated in either of the two experimental cars, and the other was operated in one of the standard cars in the test train. The surfaces on which the blocks were mounted were leveled carefully for each set-up. The test cars consistently showed fewer "tumbles" than did any of several standard-type cars used on various test runs. A typical record is given:

Number of Falls Recorded on a Run of 85 Miles at an Average Speed of 61 mph

Lateral acceleration (% of gravity)	9%	12.6%	19%
Test Car A-22	170	20	0
Modern Standard Car	582	54	4

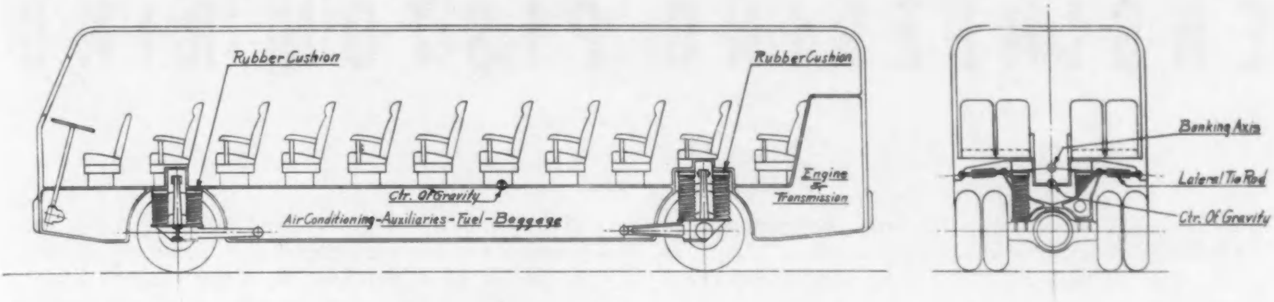
The degree of accuracy of the tumble-block records is indicated by a close correlation with data taken from an entirely different source. The number of deflections of the lateral-control arms of several selected magnitudes were recorded for a known mileage. Since the spring rate of this unit was known, these deflections were converted to forces, and thus force expressed as a percentage of the body weight was plotted against number of occurrences per mile. This curve, Fig. 8, agreed closely with the results of the tumble blocks.

The initial test runs of the steel cars were observed by some who feared that the necessary differences in weights and sizes of all parts might have introduced some detrimental factors in the riding performance not present in the experimental cars; however, it is gratifying to all believers in sound fundamental engineering principles that, when these principles are strictly adhered to, consistently satisfactory results are achieved. The same superior riding qualities were experienced with the steel cars from the very beginning that were so marked in the road-test program with the experimental cars.

A large number of test runs and six months' service have shown that the new pendulum cars are unusually quiet and that the riding qualities represent a distinct improvement over existing modern equipment. Although this type of car rides well when coupled to standard cars, the best riding qualities and full action of the pendulum suspension can be experienced only when the car is coupled between other cars of similar design. Since the banking of the pendulum car is opposite in direction to the roll of the standard car, there is more relative movement and greater forces acting at the diaphragm when the pendulum car is coupled to a standard car than when coupled between other pendulum cars. While these forces restrict the pendulum action and tend to introduce some shock and vibration into the car, the generally excellent riding qualities are apparent to passengers comparing this car with other cars, even when the car is coupled between others of standard type.

Accelerometers recording lateral and vertical shocks, as designed, built, and operated by the Santa Fe Test Department, gave consistent quantitative data throughout the experimental car and steel car test programs, showing fewer shocks and better riding qualities for the pendulum cars than for any conventional cars ever used in the test trains. Both conventional heavyweight, and modern lightweight cars of the latest design were used for comparison.

A survey covering several runs was made by one of the interested railroads operating these pendulum cars in regular service in a train composed of modern streamliner equipment. Of 195 passengers interviewed, 90% answered "yes" to the question: "Is there any noticeable difference



■ Fig. 10 – Diagrammatic application of pendulum suspension to motorcoach

in riding qualities?," this question, of course, being directed to those passengers who were given an opportunity to experience riding conditions in the pendulum car and in other cars on the train. Every passenger interviewed commented favorably upon lack of vibration, soft and smooth gliding action of car, and absence of noise and rumble when the car was in motion. Nearly all automotive engineers who have been responsible for working out the "rides" on new models have, when the final selection was to be made, resorted to the evidence of the "seat" test rather than reliance upon the voluminous data taken from ride instruments.

There is no question that the great improvement in riding qualities in American passenger automobiles since the advent of independent suspension and soft springs in 1934 has made a very favorable impression on the buying public. The further improvements through the use of frictionless coil springs all around on a number of cars, together with the almost complete abandonment of leaf springs on all modern railway passenger trucks, suggest serious consideration of coil springs for motor coaches. Good isolation from road shocks and vibration is achieved by a combination of larger static deflection, frictionless springs, and liberal use of soft rubber in series with the springs. Of course, adequate hydraulic control is essential.

The narrow spring spacing on a bus with large dual tires does not permit adequate static deflection unless the body is stabilized to prevent excessive roll on even moderate curves. It is realized that static deflection is limited also by height variations due to the relatively large weight changes. However, it is believed that, if larger spring deflections are included in the basic design, height variations will not cause objectionable conditions.

Fig. 10 is a schematic presentation of one possible arrangement of springs and lateral control parts that would permit soft springs and achieve also a pendulum-like action on curves rather than the present roll which is at times frightening to passengers who are not aware of the wide, safe tread on the modern bus and its actual great resistance to overturn.

A bus, not being restrained laterally by flanged wheels to a track, is not subject to lateral shocks of any magnitude; however, it is probable that a slight amount of cushioning would be desirable in the lateral tie-rods as indicated in the diagram.

The pendulum suspension applied to railroad cars is simpler than conventional suspensions and has no moving surfaces carrying the body weight. The same mechanical advantages are believed possible on a motor bus.

A freely resilient support – inherently stable – can make a motor coach as comfortable-riding as any vehicle on the highway or on rails.

Diesel Lubricating Problems

It is well known that the modern high-speed diesel engine, as such, is more subject to engine deposits, greater sludge formation in the lubricating oil, and engine wear than automotive gasoline engines. The problem of satisfactory lubrication is further complicated by the variables of engine design and operation. These variables largely determine upon which requirement of the lubricant the emphasis shall be placed. The intensive work done by those interested in these problems, the technical men and engineers in both the petroleum and engine fields, has resulted in remarkable progress during the past few years. One only has to look at a picture of a typical piston from an engine using what was considered a satisfactory lubricant a few years ago and compare it with a picture of a piston from an engine using one of the lubricants available today, to visualize this progress. The improvement shown by this comparison has been obtained by intensive studies of mineral oils, additives and their various combinations.

Engine deposits in the modern high-speed diesel engine are of particular importance. This phase of diesel lubrication has been the subject of many investigations for, unless deposits are reduced to a minimum, faulty engine operation will occur within a relatively short period of service. While today, the function of additives cannot be overlooked in connection with this problem of engine deposits, the properties of the mineral oils are equally important. A more complete knowledge of the characteristics of mineral oils is essential for the development of a satisfactory lubricant.

While the general characteristics of lubricating oils obtained from the various classes of crudes are rather well established, the relative merits of these crudes and their refining treatment is the subject of considerable discussion.

Excerpts from the paper of the same title by A. E. Smith and J. P. Stewart, Socony-Vacuum Oil Co., Inc., presented at the National Fuels and Lubricants Meeting of the Society, Tulsa, Okla., Oct. 23, 1941.

ENGINEERING PISTON RINGS

THE words "high-speed diesels" have normally such a broad interpretation that it is wise before talking of piston rings for such engines to state what is considered "high speed."

If these engines are classified according to Maleev's speed factor, we can select those of over 27. This classification includes engines up to 7-in. stroke running over 1500 rpm (Fig. 1) and, assuming that maximum bore would be a square engine, this would mean engines of less than 7-in. bore. This classification takes in all automotive-type diesel engines and many of the high-speed marine engines. All of these engines also have one other thing in common, and that is the piston is lubricated from the crankcase oil and not by lubricators.

[This paper was presented at the ASME 15th National Oil and Gas Conference, the SAE Diesel-Engine Activity cooperating, Peoria, Ill., June 18, 1942.]

The difference to be considered regarding piston rings engineered for low-speed diesels and high-speed diesels is that, due to the higher rpm, all of the inherent limiting characteristics of a ring are exaggerated and the tendency to pump oil is increased. The curve (Fig. 1) gives an idea of the range where this increased rpm becomes effective. Above the speed-factor line of 9 and below 27, difficulty of oil control may be noticed. Above the 27 line, it is pronounced.

■ Piston and Ring Assembly

The piston rings function primarily in such a way as to make the piston capable of performing its purpose of transmitting power with a minimum amount of deterioration; secondly, they permit the heat flow of the products of combustion to the cooling medium through the unit of the piston and the rings; and thirdly, the rings control the

RING-STICKING, broken rings, and excessive wear, the most usual sources of complaint on high-speed diesel engines, have been improved greatly in many cases by using some of the excellent oils now offered by many oil companies, Messrs. Lane and Nixon aver in a summary of their recommendations to improve the performance of diesel piston rings.

In the case of broken rings, they point out, breaks at the back are usually the result of excessive ring loading or poor design. Tip or end breakage (commonly called fatigue breakage), assuming clearances are correct, is usually the result of flutter or ring vibrations. One of the causes of this flutter, they explain, is the high pressure rise during combustion, particularly in a cold engine. Change of ring proportions, even use of higher tensions, they reported, often will eliminate this complaint.

Checking and correction of injection timing often will stop the breakage. An engine running at uniform speed and load has far less breakage, they point out, than has one operating under conditions of frequent starting and stopping. Rings do not break up all at once, they emphasize; fracturing of rings into small pieces is only the result of running after the initial failure.

Summarizing their remarks on ring wear, the authors report:

"Ring wear normally increases with speed. Excessive wear is, in most cases, a result of dirt getting into the engine and, where particularly dusty conditions are encountered, frequent drainage and proper maintenance of air and oil filters retard wear. Engine maintenance, including the fuel and injection system, is important. In most cases, maximum oil economy and minimum wear do not go hand in hand. Usually, it is more economical to drain the oil at proper intervals.

"When conditions are unusually severe, the application of surface-treated, hard-plated, or metal insert rings frequently lengthens the period between overhauls. Initial scuffing and break-in wear are factors of cylinder finish and ring surfaces, together with other variables requiring special consideration for each type of engine."

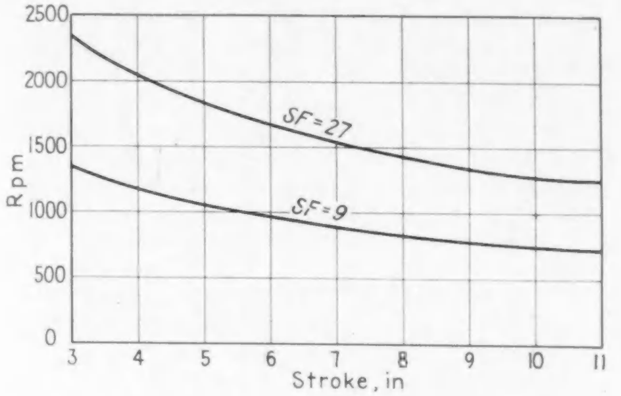
★ ★ ★

THE AUTHORS: PAUL S. LANE (M '36) became research engineer with the Muskegon Piston Ring Co. in 1940, after having been chief metallurgist for five years at the American Hammered Piston Ring Co. He graduated in 1922 from the Baltimore Polytechnic Institute, and continued his technical studies at night classes of the Johns Hopkins University. Mr. Lane has presented technical papers dealing with piston rings and the wear of gray cast iron in aero, diesel, and automotive engines before several engineering societies. STUART NIXON (M '32) graduated from the Massachusetts Institute of Technology and joined the Continental Motors Corp. as production engineer; he was successively sales engineer for industrial sales and Western District manager. He then became a member of the engineering staff of the Sealed Power Corp. in the capacity of sales engineer, the position which he now holds.

for High-Speed Diesels

by PAUL S. LANE
Muskegon Piston Ring Co.

and STUART NIXON
Sealed Power Corp.



■ **Fig. 1 – Speed factor versus stroke and rpm – SF = N^2L (1.6); N = rpm in thousands; L = stroke in in.**

lubricating oil – preventing it from getting into the combustion chamber in excess of that required for lubrication.

In considering the piston ring set-up, it is essential to consider the *number* of rings per piston, and the *location* of these rings (see Fig. 2 showing various arrangements of rings in five piston designs). The heat reservoired in the piston and the time interval for its flow, therefore, control the number of piston rings required.

The number of rings affects the average piston head temperature, for it is through the rings that most of the heat is conducted to the cylinder walls and to the cooling medium. The first ring takes the majority of the heat, and the second ring will take about one-half of the same amount, and the third ring about a quarter. Now, if the temperature drop is enough, then no more rings are required. It has been shown[1] that piston rings can affect the head temperature about 35 F but, after the optimum number of rings are in the belt, the addition of more rings does not affect the piston temperature sufficiently to measure. However, as the top ring begins to fail, the thermal load placed on the lower rings is increased and sufficient rings should be used to take care of this condition. In general, a minimum of three compression rings is required.

The location of the piston rings is of great importance. If the top ring is too close to the head of the piston, trouble will be experienced from the ring operating too hot, resulting in a stuck ring, loss of tension, and rapid wear. If located too far below the piston head, the hot gases will destroy the top cylinder lubrication and will not permit a satisfactory heat flow to the walls. A top land width of 0.125 in. per in. of diameter up to 0.175 in. per in. of diameter is quite commonly used for lands between compression rings. Ring lands should never be narrower than the ring and, under most circumstances, approximately 1/20th to 1/30th of the bore diameter minimum. The oil control rings are preferably located above the piston pin, to permit sufficient lubrication of the gliding surface of the piston. Where we have to contend with a high rod throw-off, it is sometimes advisable to use a ring in the skirt as a metering ring to reduce the amount of oil that the oil control ring above the pin has to handle.

The piston fit is a very important factor in aiding oil control and heat dissipation. The closer the piston fit, the better the oil and heat control will be. The utilization of cam grinding of pistons to prevent the scoring of pistons when fitted closely is becoming a regular practice today.

Piston stability deserves more consideration than it is getting. The tendency is to shorten pistons and rods and, where we do, there is more angularity and more rocking of the piston. This condition results in the tendency to make the rings leave the wall. Rings can be located and designed to have a stabilizing effect which is an important consideration in this design.

Aluminum pistons will run cooler than cast-iron pistons[2] so that the thermal load on piston rings is greater with cast-iron pistons than aluminum pistons, which results in an increase in the requirements for cooling these pistons. Utilizing the lubricating oil for cooling the pistons under the heat is often the distinguishing item between a successful ring life and an abnormal failure.

■ Factors Affecting Oil Consumption

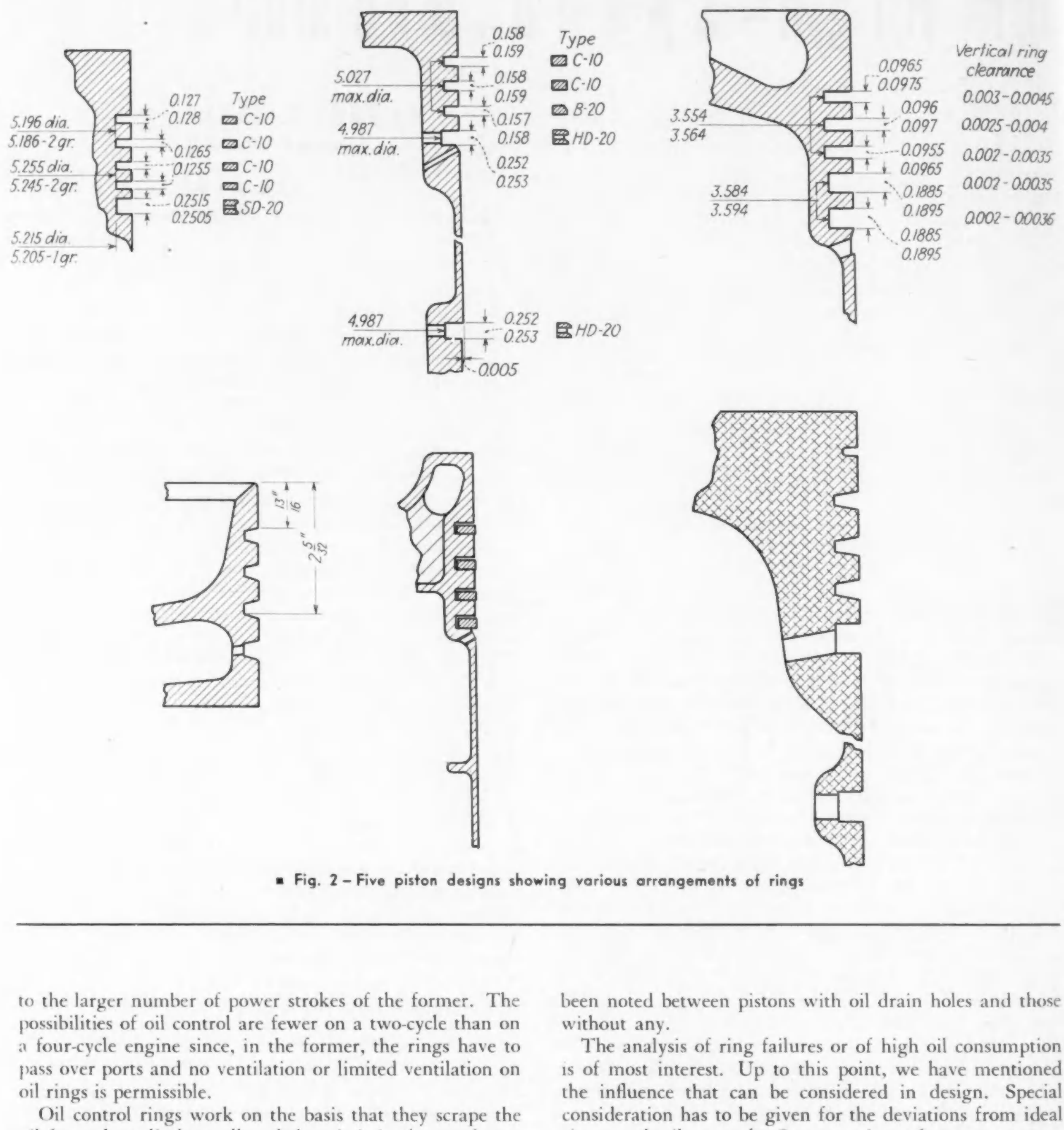
The gliding area of the piston is supposed to run on an oil film. A certain amount of this film is absorbed by the liner, and the rest adsorbed. This film on the power stroke is subjected to the heat of combustion on the outer layer and some of the adsorbed layer is burnt off. The cooling medium holds the inner layer down below the fire point, and that, plus the insulating property of the oil, prevents a complete burning off. Some of the adsorbed oil is also released to replenish this oil to keep a balance. Since the thickness of this film is in the nature of 1000 Angstrom units[3] and, if oil balance is kept perfect, oil consumption is a very small amount per thousand rpm. However, such perfect control cannot be maintained, and an excess of oil must be supplied which is itself burnt in the combustion. It is the control of this excess that is the measure of ring performance.

The oil consumption on two-cycle engines is usually accepted at a higher figure than on four-cycle engines, due

[1] See contribution of C. G. A. Rosen to "Symposium on Piston Temperatures," presented at the Annual Meeting of the Society, Jan. 13, 1939.

[2] See Purdue Engineering Experimental Station Bulletin No. 25: "Flow of Heat in Pistons," by Huebottom and Young.

[3] See ASME Transactions, Vol. 61, 1939, pp. 633-641: "Studies in Boundary Lubrication," by W. E. Campbell.



■ **Fig. 2 – Five piston designs showing various arrangements of rings**

to the larger number of power strokes of the former. The possibilities of oil control are fewer on a two-cycle than on a four-cycle engine since, in the former, the rings have to pass over ports and no ventilation or limited ventilation on oil rings is permissible.

Oil control rings work on the basis that they scrape the oil from the cylinder wall and deposit it in the crankcase. Oil drainage through drilled holes or slots on the piston-ring groove is the most common method. This method is usable with all types of rings to an advantage. The main requirement is to have ample size holes and drainage area. Drainage below the oil ring land is effective and sometimes even both drainages are used. The other resort for drainage is relief of the piston. For example, if a ring is used in the bottom of the skirt, grinding undersize the lower edge below the ring groove provides all the drainage needed. This condition applies up to the point where the pressure of the column of oil is equal to the oil pressure created by the ring. On loose-fitting pistons of small length over diameter ratio, no difference of oil consumption has been noted between pistons with oil drain holes and those without any.

The analysis of ring failures or of high oil consumption is of most interest. Up to this point, we have mentioned the influence that can be considered in design. Special consideration has to be given for the deviations from ideal rings and oil control. Some engines, for no apparent reason, are oil-burners. To take care of these conditions, oil rings of various degrees of drasticity are available. If we start the list with the plain compression ring of normal tension, the other end is a flexible high-unit-pressure narrow-contact ring.

■ Ring Types

Ring types include the plain ring of rectangular cross section, the edge scraping ring, the high-unit-pressure scraping ring, the oil control rings, which include the ventilated, the scraper type, the multiple piece rings, the steel segment rings, and the scraper type rings. (See Fig. 3, left.) There are a few special types, such as the taper-side

rings, the metal insert rings, and the multiple-grooved rings, which have special applications. The general rule always to follow is to use the simplest piston ring which will perform satisfactorily.

■ Tension and Unit Pressure

The plain compression ring is the basic ring for all compression and scraper rings. It has certain characteristics built into it that are varied for the particular application. The tension and unit pressure are important. In general, the higher the rpm, the greater the unit pressure required up to the maximum attainable without overstressing the ring. The unit pressure is a function of the diameter-over-thickness ratio and the free-opening-over-thickness ratio. So, by varying these proportions, it may be increased or decreased as desired. Unit pressures run up to 45 psi on plain compression rings. The pressure created by the ring itself is very low compared with the pressure behind the upper rings due to the gas pressure[4], and so the common misunderstanding that high unit pressure means high friction losses is not substantiated, except on very low speed within the cranking range. Narrow-land, single-piece oil rings with pressures up to 100 psi are manufactured and, on spring-supported oil rings, pressures as high as 250 psi have justified their application and proved successful.

■ Ring Widths

The trend for the past number of years has been to make use of narrower compression rings. The prevailing ratio (diameter-to-width) of gasoline automotive engines has been from 22 to 45 up to 3½-in. bores with the trend towards the higher figure rather than the lower figure. On high-speed diesels with bores of 3½ to 10 in., the range has been running 35 to 65. The engineers who have worked toward the narrower rings report a decided improvement in performance. This improvement is traceable in the reduction of land wear resulting from the lower inertia, less wear of both the cylinder and rings, better ring arrangements; that is, less piston space occupied by the piston rings. Decreased friction and a far better sealing action, and the elimination of ring flutter at high speeds also result.

A graphical picture of the effect of ring width on the natural period of the ring is illustrated by the graph of rpm (Fig. 4) at which a break in blowby of an internal-combustion engine occurs.[5]

■ Compression Rings

Compression rings are primarily to control blowby, but they do affect the oil consumption. To control blowby, the ring must stay in contact with the cylinder wall, and not go into a flutter. Damping characteristics can be built into a ring by incorporating twist. After its primary function of controlling blowby, the secondary functions of the compression rings are to dissipate heat and spread the remaining oil uniformly. The quantity of heat flowing into and out of a ring does not seem to be a function of its width. It does have some relation to the radial thickness, and, on high-output engines the use of narrow *high-wall* rings,[6] has proved beneficial.

[4] See *Diesel Power*, Vol. XII, No. 10, October, 1934, pp. 548-555: "Pressure behind Piston Rings," by A. J. Robertson and A. R. Ford.
[5] See *Automotive Industries*, Vol. 84, No. 7, April 1, 1941, pp. 386-387: "Ring Width and Blowby," by J. H. Ballard.
[6] See SAE Handbook, 1942 Edition, pp. 54-61.

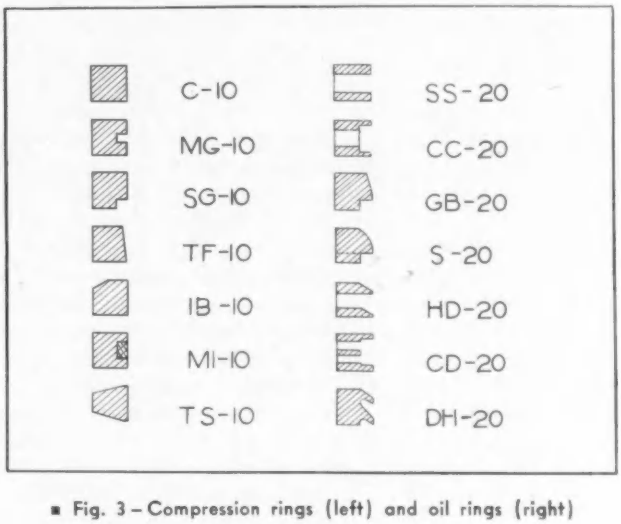
■ Oil Rings

Oil rings according to degree of scraping action are as follows (Fig. 3, right):

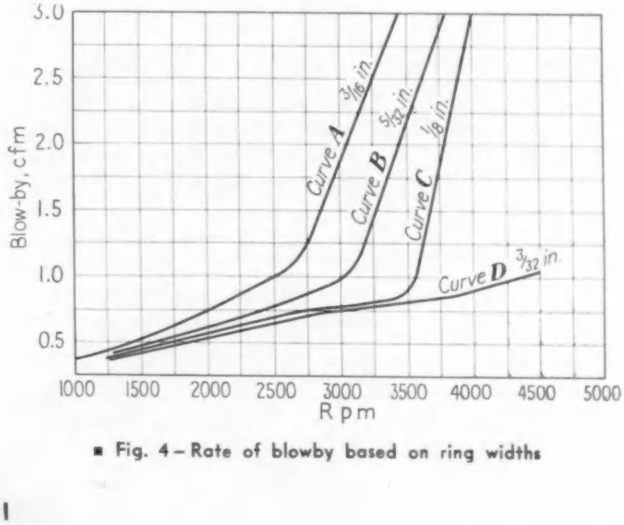
(1) Plain compression
(2) Twisted and taper-faced rings
(3) Beveled rings with narrow lands
(4) Beveled rings with narrow lands with oil reliefs on underside
(5) Single slot ventilated wide lands
(6) Channelled and slotted rings
(7) Single-slot narrow lands (beveled lands)
(8) Nos. 5, 6, & 7 spring-supported
(9) Segmental rings with spring supports
(10) Steel or bronze ribbon-type

Designs 1 to 5 inclusive have their broadest applications in the slower-speed engines. Nos. 6 to 10 inclusive are regularly used in engines operating in the higher speed ranges. Particular attention is called to Item 8, referring to Types 5, 6, and 7 as used with a steel expander ring to furnish increased tension. While expander or spring-supported type rings received only limited application in diesel engines of a few years ago, they more recently have proved successful when properly engineered and applied.

In order to use expander-type rings successfully, consideration must be given to groove depth, and to the radial



■ Fig. 3 – Compression rings (left) and oil rings (right)



■ Fig. 4 – Rate of blowby based on ring widths

wall thickness of the cast-iron ring. Best results are obtained only when the expander and ring assembly are designed specifically for each piston and engine.

Again in the case of side clearance, experience has shown that this clearance should be increased over the clearance used with an unsupported ring. Spring-supported rings have proved beneficial as a result of their higher tension, their ability to conform to cylinder shape due to their flexibility and, as a result of their action, they promote piston stabilization. Experience has shown that such rings, including those having steel segments, do not cause increased cylinder or ring wear.

Piston-Ring Troubles

The failure of piston rings to perform satisfactorily is usually first noticeable by increased oil consumption. When increased oil usage, chargeable to the rings themselves, is experienced, the following difficulties can be looked for:

Cause	Solution
1. Sludge blocking off oil rings.	1. Eliminate cause of sludging. Change to different grade of oil or drain oil more frequently. Improve crankcase ventilation. Use slotted oil rings with maximum slot width or change to scraper-type ring, possibly including an alteration in groove drainage.
2. Compression rings failing or sticking.	2. Change set-up to allow additional lubrication to compression rings. This change also improves cooling. Change type of oil to alter coking zone. Determine if piston lands are distorted and pinching the rings. Improve heat dissipation by a change in piston-head design or through the use of additional compression rings.
3. Piston scuffing resulting in ring damage.	3. Eliminate cause of scuffing by changing piston shape or clearances. In some cases a less severe skirt ring will prove beneficial. Grade and quality of oil are also factors.
4. Lacquer and varnish formations tending to stick rings.	4. Check heat zones and reduce piston and ring temperatures as far as possible. Also check clearances of both piston and rings. Special-type oils ofttimes give relief from this trouble. Crankcase temperature and ventilation are factors.
5. Excessive wear.	5. Check maintenance of air and oil filters. Alter piston-ring oil-control set-up to permit more adequate lubrication to piston skirt and to upper ring belt.
6. Broken rings.	6. Check side gap, and groove clearance. Check injection timing. Breakage can be caused from excessive side wear of either piston rings or lands. Breakage also can be caused by insufficient side clearance which causes stuck rings. See the following:

Too little side clearance	Too much side clearance
1. Ring-sticking.	1. Groove pounding.
a. High blowby.	a. Land wear.
b. Rapid ring wear.	b. Increased blowby.
c. High oil consumption.	c. Ring breakage.
d. Ring breakage.	
Too little gap clearance	Too much gap clearance
1. Ring ends butting.	1. Leakage.
a. Broken rings.	a. High blowby.
	b. High oil consumption.
	c. High rate of wear.
Too little groove clearance	Too much groove clearance
1. Insufficient room for carbon build-up.	1. Increased leakage.
2. Broken rings.	2. Broken piston lands.

General

Ring sticking, broken rings, and excessive wear are the most usual sources of complaint on high-speed diesel engines. In many cases these troubles have been improved greatly by using some of the excellent oils now offered by many oil companies. In the case of broken rings, those that break at the back are usually the result of excessive ring loading or poor design. Tip or end breakage, assuming that clearances are correct, is usually the result of flutter or ring vibrations, and is commonly called fatigue breakage. One of the causes of this flutter is the high pressure rise during combustion, particularly in a cold engine. Change of ring proportions, even higher tensions, often will eliminate this complaint. As mentioned previously, injection timing will stop the breakage. An engine running at uniform speed and load has far less breakage than one operating under conditions of frequent starting and stopping. Rings do not break up all at once. Fracturing of rings into small pieces is only the result of running after the initial failure.

Ring wear normally increases with speed. Excessive wear is, in most cases, a result of dirt getting into the engine and, where particularly dusty conditions are encountered, frequent drainage and proper maintenance of air and oil filters retard wear. Engine maintenance, including the fuel and injection system, is important. In most cases, maximum oil economy and minimum wear do not go hand in hand. Usually it is more economical to drain the oil at proper intervals.

When conditions are unusually severe, the application of surface-treated, hard-plated, or metal insert rings frequently lengthens the period between overhauls. Initial scuffing and/or break-in wear are factors of cylinder finish and ring surfaces, together with other variables requiring special consideration for each type of engine.

Conclusion

In this paper we have attempted to give some general recommendations to improve the performance of diesel engine piston rings. Some of the many factors which affect ring life, and particularly oil control, have been discussed briefly. As indicated by the title of the paper, optimum results for any engine are obtained when rings are engineered specifically for the job.

This work has been prepared as part of the Symposium on the Control of Lubricating-Oil Consumption in High-Speed Diesel Engines and, as such, it is hoped that the data submitted will prove helpful in outlining some of the piston-ring problems connected with this subject.

BEARINGS and BEARING CORROSION

by LEONARD RAYMOND

Supervisor, Automotive Laboratory, Tide Water Associated Oil Co.

DATA presented in this paper indicate the bearing corrosion problem to be fairly straightforward in that:

1. Corrosion of copper-lead bearings specifically can be reduced by improving the fineness of the microstructure;

2. All bearings appear to be aided greatly by reduction of operating temperatures; and

3. Treating the corrodible bearings by special processes, such as indium plating, greatly increases their corrosion resistance.

However, anomalies are cited which indicate that bearing and oil combinations do not behave in relatively the same manner in all engines.

Different types of corrosion may occur in different engines or bearings, or a mechanical or assembly defect, rather than a corrosive oil, may be responsible for bearing failure. The appearance of a bearing frequently fails to indicate the reason for its failure, and more thorough investigation must be made.

In his conclusion the author emphasizes that the field of bearing corrosion still has large unexplored areas.

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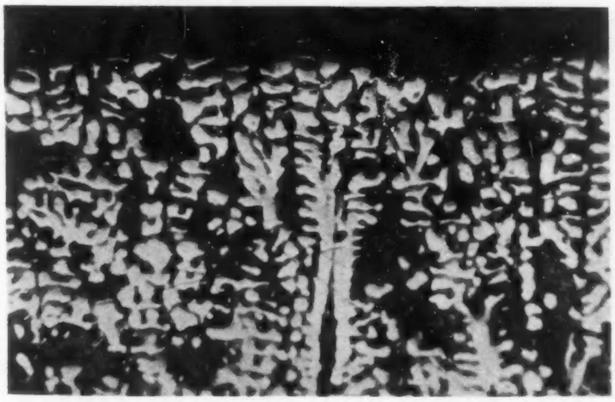
"BABBITT" bearings have been accepted generally as the preferred bearing metal for by far the great majority of bearing applications due to their low friction characteristics, conformability, embeddability, bonding characteristics, and corrosion resistance.[1] However, under conditions of increased bearing loads and higher oil temperatures, babbitt bearings of conventional design are susceptible to fatigue failure, resulting in breaking out of the bearing metal. In such cases, protection has been sought in bearing metals with improved high-temperature strength characteristics, the bearing alloys resorted to being copper-lead, cadmium-silver, cadmium-nickel, and hardened high lead (98% lead), as well as other high-lead alloys. The most widely used of these newer types of precision bearings has been the copper-lead. A typical proximate analysis of this bearing metal is 65% copper, 35% lead. Due to the mutual insolubility of lead and copper, the structure consists of lead distributed throughout a dendritic copper matrix, this bearing mixture being somewhat unusual in



■ Fig. 1 – Coarse-structure copper-lead bearing (125X magnification)

that the matrix is the harder constituent. The dimensions of the copper dendrites and the degree of lead segregation range from a relatively coarse structure as shown in Fig. 1, to a relatively fine structure, Fig. 2, depending upon the manufacturing process. The white constituent is copper, the gray lead.

In common with the other higher-strength bearing alloys just mentioned, the copper-lead bearings lack the chemical

[This paper was presented at the National Fuels and Lubricants Meeting of the Society, Tulsa, Okla., Oct. 23, 1942.]

[1] See *Industrial and Engineering Chemistry*, News Edition, Vol. 14, No. 21, Nov. 10, 1936, pp. 425-428: "The Newer Bearing Materials and Their Lubrication," by H. C. Mougey.

ineitness of the babbitt bearings, being corrodible by products of oil oxidation. This corrosion attack, in most cases, results in solution of the lead phase and production of a porous and weakened structure of copper. If sufficient lead is removed, the porous copper is crushed easily or its bond to the backing metal is lost, leaving large sections of lead-free copper which may be detached readily from the steel support.

The behavior of copper-lead bearings in the field has been erratic in many cases, and it was believed that some knowledge of the critical operating factors was very desirable. The metallurgical construction of the copper-lead bearing led to the opinion that the degree of lead segregation or fineness of structure might affect the susceptibility to penetration and attack by the corrosive materials. If this opinion were found to be correct, a means of improving corrosion resistance would be indicated.

The recent work by this laboratory on the comparison of coarse- and fine-structure copper-lead bearings was carried out in a 3-cyl, high-speed, two-stroke-cycle diesel engine operated at 82 bhp at 2000 rpm. This engine employs oil cooling to maintain proper piston temperatures, and the oil temperatures consequently are relatively high despite the use of an oil cooler. The bearings used as standard equipment are coarse structure; the fine-structure bearings were specially manufactured for this investgation. Engine tests of 500-hr duration in which one standard coarse-structure connecting-rod bearing was replaced by a fine-structure bearing were made at a controlled oil base temperature of 230 F. A variety of commercial and experimental oils were run, the results being reported in Table 1 as follows:

Table 1 – Comparison of Coarse- and Fine-Structure Copper-Lead Bearings in 500-Hr Runs at 230 F Oil Base Temperature (Weight Losses* in G/Shell)

OIL	Connecting Rod* Coarse	Connecting Rod* Fine	Main* Coarse	Main* Fine
Oil A, Commercial, Non-Additive	0.091	0.095	.....	.....
Oil A plus Additive I (Inhibitor, Detergent)	.....	.....	0.203	0.092
Oil B plus Additive II (Inhibitor, Detergent)	0.047	0.090	.....	.....
Oi C, Commercial, Additive	0.050	0.060	.....	.....
Oil D, Commercial, Additive	0.168	0.095	.....	.....

* Connecting-rod losses for upper shell only; main bearing losses for lower shell.

The data show little difference in the relative corrodibility of the two structure types at low levels of weight loss, although there is a slight advantage for the fine structure. The constancy of weight losses with the fine-structure bearings is noteworthy, the values apparently being normal wear.

All of the oils tabulated are substantially non-corrosive under the test conditions, and it would be expected that no service troubles should be encountered with them.

However, it has become apparent from the recent exchange of experiences of bus and truck operators and oil refiners, that bearing failures, not confined to one oil or one type of service, have been occurring with similar oils.[2] Analysis of some failures has indicated assembly defects or excessive oil temperatures resulting from impaired cooling to be the principal offending factors. With higher engine

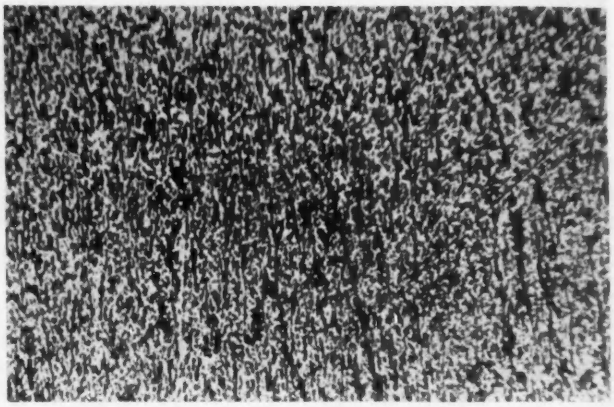


Fig. 2 – Fine-structure copper-lead bearing (125X magnification)

outputs accompanied by cooling limitations apparently becoming more probable, corrosion difficulties due to excessive oil temperatures became of increasing concern. In view of these factors, a series of runs of 100 hr duration at 260 F oil base temperature without other change in operating conditions were made with the engines. The bearing weight losses for the coarse- and fine-structure bearings are summarized in Table 2:

Table 2 – Comparison of Coarse- and Fine-Structure Copper-Lead Bearings in 100-Hr Runs at 260 F Oil Base Temperature (Weight Losses* in G/Shell)

OIL	Connecting Rod* Coarse	Connecting Rod* Fine	Main* Coarse	Main* Fine
Oil A plus Additive I (Inhibitor, Detergent)	0.837	0.046	0.594	0.030
Oil A plus Additive I' (Inhibitor, Detergent) Higher Concentration	0.383	0.088	.....	.....
Oil B, Non-Additive	5.755	2.740	.....	.....
Oil B plus Additive I (Inhibitor, Detergent)	.....	.....	4.595	0.153
Oil B plus Additive III (Inhibitor)	1.650	1.037	.....	.....
Oil E plus Additive I (Inhibitor, Detergent)	3.518	0.268	8.077	2.186
Oil E plus Additive IV (Inhibitor)	5.084	2.661	8.904	2.740
Oil C, Commercial, Additive	8.596	3.061	.....	.....

* Connecting-rod losses for upper shell only; main bearing losses for lower shell.

The data demonstrate definitely that the fine-structure copper-lead bearings are much less susceptible to corrosion than are the coarse-structure bearings. The surface appearance of the fine-structure bearings after use was quite different from the coarse bearings, the fine-structure surface being a dull brown while the coarse-structure bearings had a bright coppery appearance. A preferential loss of lead, shown by photomicrographs, accompanied the coppery color. It should be mentioned that this burnished surface is not specific to corrosive removal of lead, since the same surface appearance has been observed in the field in the absence of corrosion and apparently due to metal-to-metal rubbing.

In Table 3, the results of the 500-hr runs at 230 F and the 100-hr runs at 260 F oil temperature are compared for the coarse-structure bearings.

[2] See American Transit Association Forum on Bus Fuels and Lubricants, Detroit, Mich., Feb. 17-18, 1941.

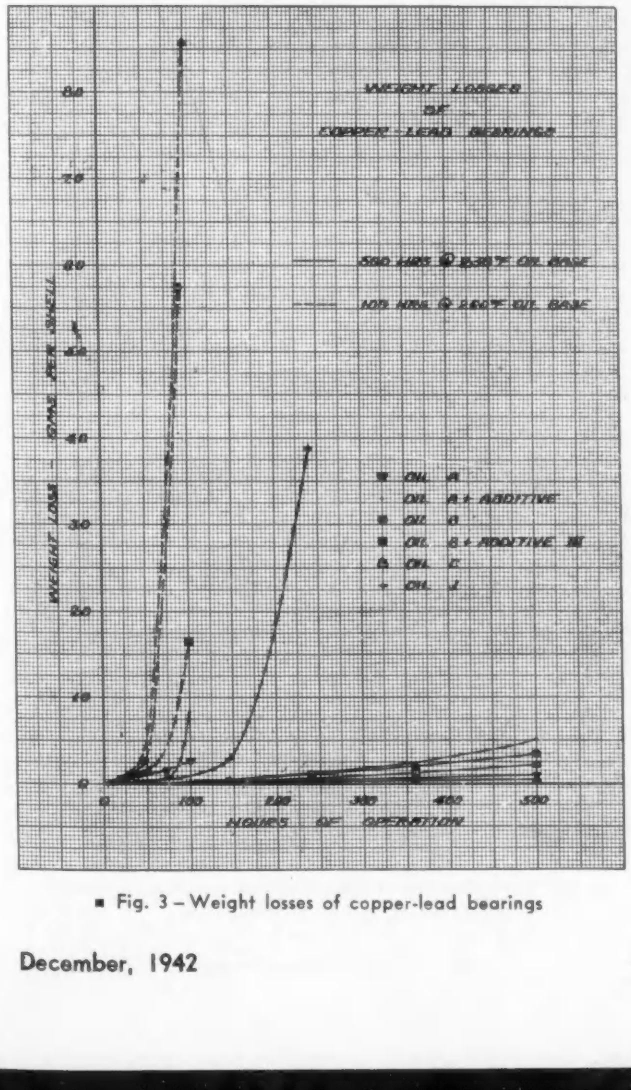
Table 3 – Comparison of 500-Hr Runs at 230 F Oil Base Temperature With 100-Hr Runs at 260 F Oil Base Temperature – Coarse-Structure Copper-Lead Bearings Only (Weight Losses* in G/Shell)

OIL	Length of Run : Temperature	Connecting Rod* 500 Hr 230 F	Connecting Rod* 100 Hr 260 F	Main* 500 Hr 230 F	Main* 100 Hr 260 F
Oil A, Commercial, Non-Additive		0.091	0.270	0.066	1.031
Oil A plus Additive I (Inhibitor, Detergent)		0.505	0.837	0.203	0.594
Oil B, Non-Additive		0.355	5.755	0.244	4.861
Oil B plus Additive III (Inhibitor)		0.216	1.650	0.192	3.610
Oil C, Commercial, Additive		0.050	8.596	0.055	8.129

* Connecting-rod losses for upper shell only; main bearing losses for lower shell.

Study of the data indicates that the rise of 30 F in oil base temperature is much more destructive in its corrosion effects than a five-fold increase in operating time. The appreciable effect of fine structure in reducing relative susceptibility to corrosion should be emphasized, since the results demonstrate that the replacement of coarse by fine structure bearings in an engine operating in the critical range on available oils will raise the corrosion resistance to a safer level. It should be pointed out also that the finer structure has been found to have better strength characteristics,[3] which are particularly important with increasing temperatures.

The critical effect of the increase in temperature from 230 F to 260 F applies in a greater or less degree to all of the oils in Table 3. However, the critical corrosion temperature under any specified conditions of operation will vary with different oils. Thus a number of the lubricants tested in the course of this work were found to be very corrosive at 230 F, their critical temperature range being some lower level.



■ Fig. 3 – Weight losses of copper-lead bearings

The differences in temperature sensitivity of the oils as regards corrosion should be noted. Oil C, which showed the lowest weight losses at 230 F, was the most corrosive at 260 F. Oil A plus Additive I gave somewhat higher weight losses than the other oils at 230 F; yet was the most satisfactory at the higher temperature.

The nature of the bearing corrosion-time curves are of interest in that the weight losses with corrosive oils show a sharp break characteristic of an induction-period effect. The weight-loss data for a number of oils have been plotted in Fig. 3 and demonstrate graphically the exponential character of the corrosion-time curve under conditions of excessive corrosion, and the detrimental temperature effect.

The great importance of temperature on bearing corrosion was noted earlier in this laboratory in work with an 8-cyl, L-head passenger-car engine which brought the bearing-corrosion problem into prominence some years ago. This engine was operated in the laboratory with cadmium-silver bearings at an oil base temperature of 300 F to simulate the more severe driving conditions. In addition, a small number of runs were made on selected oils at 280 F and 320 F, with the results given in Table 4:

Table 4 – Effect of Crankcase Temperatures on Corrosion of Cadmium-Silver Connecting-Rod Bearings in 8-Cyl Passenger-Car Engine (Weight Losses in G/Complete Bearing)

OIL	280 F	300 F	320 F
Oil F, Commercial, Non-Additive	3.244	3.983	.....
Oil F plus Additive III' (Inhibitor) Lower Concentration	0.089	1.373	3.950
Oil F plus Additive V (Inhibitor)	0.009	0.010	2.716
Oil G, Commercial, Additive	0.331	0.962	.....

At 280 F, Oil F was very corrosive and Oil G moderately corrosive under the conditions of test; at 300 F Oil G and Oil F plus Additive III gave fairly high weight losses while Oil F plus Additive V continued to be corrosion-free; at 320 F, however, even Oil F plus Additive V was severely corrosive.

Indium treated cadmium-silver bearings[4] were compared with regular cadmium-silver bearings in this engine, the data being shown in Table 5:

Table 5 – Comparison of Regular and Indium-Treated Cadmium-Silver Connecting-Rod Bearings in 8-Cyl Passenger-Car Engine at 300 F Oil Base Temperature (Weight Losses in G/Complete Bearing)

OIL	Regular Cd-Ag	Indium-Treated Cd-Ag
Oil F, Commercial, Non-Additive	4.021	0.447
Oil F plus Additive VI (Inhibitor)	0.179	0.013
Oil H, Commercial, Non-Additive	0.025	0.015
Oil J, Commercial, Additive (Detergent Only)	4.995	2.323

[3] See SAE Transactions, Vol. 32, April, 1937, pp. 153-164: "Aircraft Engine Materials," by J. B. Johnson.

[4] See Technical Publication No. 900, 1938, American Institute of Mining and Metallurgical Engineers; also AIME Transactions, Vol. 128, 1938, pp. 295-307: "Indium-Treated Bearing Materials," by C. F. Smart.

The indium treatment has effected a substantial reduction in corrosion. Indium plating apparently imparts a high degree of corrosion resistance to copper-lead bearings also and has been adopted commercially by at least one engine manufacturer.

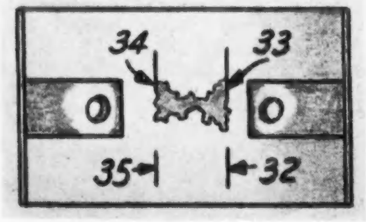
Bearings of hardened high-lead composition, containing 98% lead, were tested under conditions similar to the above, with the results in Table 6:

Table 6 – Comparison of Copper-Lead, Hardened High-Lead, and Cadmium-Silver Connecting-Rod Bearings in 8-Cyl Passenger-Car Engines at 300 F Oil Base Temperature (Weight Losses in G/Complete Bearing)

Oil	Copper-Lead	Hardened High Lead	Cadmium-Silver
Oil F, Commercial, Non-Additive	0.361	0.415	3.717
Oil F plus Additive III" (Inhibitor)	0.054	0.059	0.011
Oil F plus Additive V (Inhibitor)	0.010	0.043	0.006
Oil H, Commercial, Non-Additive	0.026	0.052	0.008

The hardened high-lead bearings showed approximately the same weight loss as the copper-lead bearings in these limited tests. It is interesting to note that, when corrosion does not occur, the cadmium-silver bearing shows lower weight losses, where, under corrosive conditions, it has been found to be much more subject to corrosion.

The data so far presented indicate the bearing corrosion problem to be fairly straightforward in that: (1) corrosion of copper-lead bearings specifically can be reduced by improving the fineness of the microstructure; (2) all bearings appear to be aided greatly by reduction of operating temperatures; and (3) treating the corrodible bearings by special processes, such as indium plating, greatly increases their corrosion resistance.



■ Fig. 4 – Connecting-rod bearing as removed in service

However, anomalies have been encountered which indicate that bearing and oil combinations do not behave in relatively the same manner in all engines. Data in Table 4 show that, in this passenger-car engine operated at 300 F oil temperature, the straight mineral oil was considerably more corrosive than Oil F plus Additive III. Tests in a Series 30 engine operated at 330 F oil base and a 1941 car engine equipped with special copper-lead connecting-rod bearings and operated at 280 F oil base confirm these results. However, when tested in a single-cylinder, four-stroke-cycle diesel engine of Comet-type combustion-chamber design operated at 10 bhp at 1200 rpm at an oil base temperature of 280 F, the additive blend gave somewhat higher weight losses than the straight oil of approximately the same base stock, as shown in Table 7:

Table 7 – Comparison of Bearing Weight Losses in Two Different Type Engines Indicating Reversal of Corrosion Effects (Weight Losses in G/Comp. Bearing)

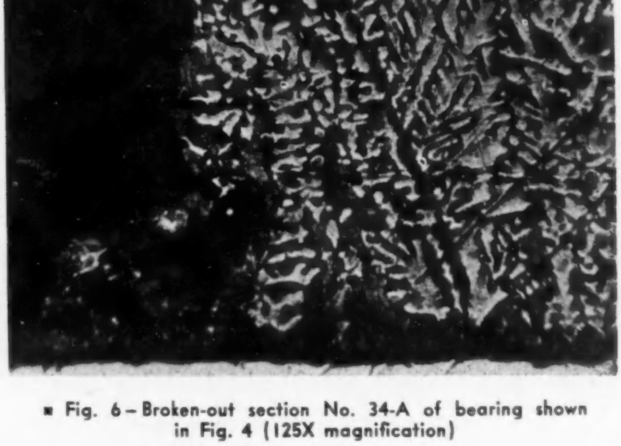
Oil	1941 6-Cyl Passenger-Car Engine, 280 F Oil Base*	Single-Cyl Comet-Type Diesel Engine 280 F Oil Line	
		First Run**	Second Run
Oil F, Commercial, Non-Additive	4.017	0.267	0.273
Oil B plus Additive III (Inhibitor)	0.233	0.426	0.397

* SAE 20 grade oil.
** SAE 30 grade oil.

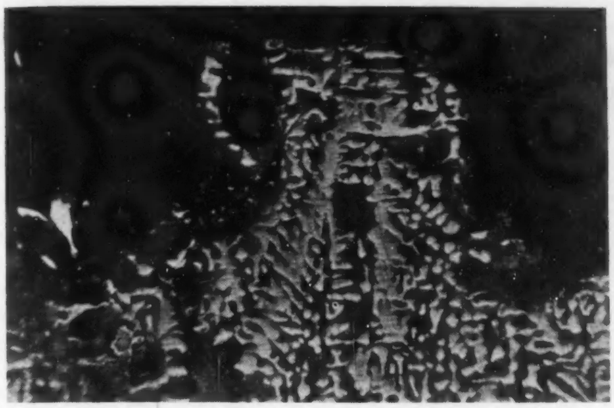
It is possible that a different type of corrosion may be occurring in this last case, in that both copper and lead may be under attack, instead of lead alone.

Reference has been made previously to the bright, coppery appearance of coarse-structure bearings corroded under accelerated temperature conditions. Fig. 4 is a sketch of a bearing removed in service due to its badly broken-out condition. This failure was originally attributed to corrosion. Photomicrographs were taken of sections through the broken-out areas and through the intact parts of the bearing, as indicated. A section through the broken-out area is shown in Fig. 5, and examination indicates that lead is present up to the edge of the cavity. Figs. 6 and 7 similarly show no loss of lead. Fig. 8, taken through an intact section, shows lead up to the surface of the bearing. The

■ Fig. 5 – Broken-out section No. 33 of bearing shown in Fig. 4 (75X magnification)



■ Fig. 6 – Broken-out section No. 34-A of bearing shown in Fig. 4 (125X magnification)



■ Fig. 7 – Broken-out section No. 34-B of bearing shown in Fig. 4 (125X magnification)

■ Fig. 8 – Intact section No. 35 of bearing shown in Fig. 4 (80X magnification)

evidence is quite conclusive that a mechanical or assembly defect, rather than a corrosive oil, was responsible for this failure. Other observations have indicated that the appearance of a bearing frequently fails to indicate the reason for its failure, and more thorough investigation must be made.

The field of bearing corrosion investigation still has large unexplored areas. There are indications that oils and bearings have different rates of response or slopes with different operating factors, including various catalytic materials. As more data on the individual operating variables become available, it may be possible to plot the response of an oil or a bearing to changes in these variables and, from the various slopes, to obtain a composite picture of change in behavior with operation factors. Until this work is done, the experimental development will, of necessity, be largely empirical.

In the meantime, it should be the aim of the engine manufacturer to decrease the burden on the oil and bearings by improved design and materials, in order to permit a greater number of oils to lubricate his engines satisfactorily; while the objective of the oil refiner will continue to be the production of lubricants more resistant to increased severity of operation. Coordinated progress of both industries is necessary for the fullest development of our automotive power.

Bibliography

1. J. K. Anthony – "Automotive Bearings," presented before the SAE Detroit Section, March 31, 1941.

2. H. L. Grange – "Metallographic Preparation of Copper-Lead Bearings," *Metal Progress,* Vol. 38, November, 1940, pp. 674-676.

3. J. N. Westsmith – "Frequent Lubrication Needed, Say Bearing Manufacturers," *National Petroleum News,* July 10, 1940, pp. 25-32.

4. C. G. Williams and H. Ludicke – "The Wear of Crankshafts with Lead-Bronze Bearings," SAE Transactions, Vol. 35, March, 1940, pp. 93-99.

5. A. B. Willi – "Engine Bearings – From Design to Maintenance," SAE Transactions, Vol. 34, December, 1939, pp. 513-525.

6. F. L. Miller – "Bearing Corrosion and Lubricants," presented before the SAE Metropolitan Section, March, 1939.

7. A. F. Underwood – "Automotive Bearing Materials and Their Application," SAE Transactions, Vol. 32, September, 1938, pp. 385-392.

8. F. G. Jenkins – "Copper-Lead Bearing Metals Favored for Severe Service," *Metal Progress,* Vol. 32, August, 1937, pp. 145-149.

9. A. F. Underwood – "Automotive Bearings – Effect of Design and Composition on Lubrication," ASTM Symposium on Motor Lubricants, March, 1937, Philadelphia, Pa.

10. C. M. Larson – "Engine Lubrication with Different Bearing Metals," SAE Transactions, Vol. 29, December, 1934, pp. 444-453.

Diesel Piston Problems

STRUCTURALLY the greatest stresses occur at the top of the piston-pin bosses, and it is necessary to provide an adequate supporting section between the pin bosses and the piston head to eliminate deflection and consequent fatigue failures. The design of the skirt construction must also be carefully reviewed to control skirt deflection. For thermal reasons the piston head and ring belt sections should be increased considerably beyond that necessary for structural purposes, in order to provide a reasonably cool piston head and ring-belt area.

With the piston structurally and thermally sound, it is necessary to determine the safe minimum clearance at which the piston skirt will operate and best results are obtained by developing a cam gradient for the piston skirt to provide the ellipticity needed for the particular engine design for minimum safe clearance and to control this ellipticity to obtain uniform skirt bearing. In our practice, the ellipticity required at the top of the skirt is determined and this ellipticity is reduced in direct relation to temperature. With the piston clearance determined, the width of the compression and oil rings should be increased to provide side clearance so that the piston in its cross-head motion at any temperature does not disturb the piston ring.

Excerpts from the paper: "Solutions for Diesel Piston Problems," by Fred Zollner, Zollner Machine Works, presented at the ASME 15th Oil and Gas Power Conference, the SAE Diesel Engine Activity Cooperating, Peoria, Ill., June 18, 1942.

AIRCRAFT-ENGINE

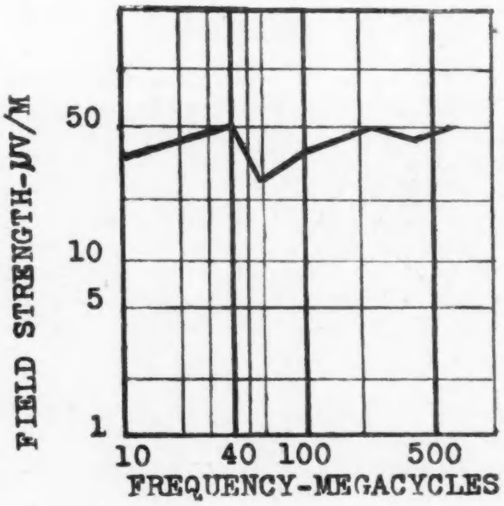
A STUDY by subcommittee E-4 of the Aeronautics Division, SAE Standards Committee, of the problem of aircraft-engine ignition shielding as it is related to the whole problem of ignition and radio shielding, was made during the preparation of Standard AS 29. Although much of this information is familiar to those who design and service such equipment, the review of the matter has brought out many points of interest and has resulted in some investigations which may be of interest to others. Hence this brief review of the problem of ignition shielding.

The reason for the existence of this rather heavy and troublesome part of the aircraft engine originates in the necessity for eliminating disturbances in the radio systems for communication and navigation used on the airplane. The disturbance produced by the high-voltage electrical system is distributed over a wide range of frequency and covers all the frequencies used in radio both of the ultra-high and moderately-high frequency ranges. The interference in moderately-high wave lengths may arise from voltage regulators, generators, and other electrical apparatus, as well as in the ignition system alone. In the short and ultra-short wave regions the ignition system is the worst offender. The disturbances that arise are very severe in the high-frequency ranges and will completely prevent radio communication unless they are eliminated. The field strength set up by a magneto system operating with unshielded leads is shown in Fig. 1. This is an approximate measurement of the horizontally polarized radiation at a distance of 100 ft from an unshielded engine. Variations in the field strength would, of course, be caused by engine mounts, cowling, and so on, but the data only serve to indicate the order of magnitude of the interference and its wide range of frequency. The use of damping resistors in the high-voltage circuits is only a partial solution to the problem and is of little value with high-sensitivity sets. To confine the field caused by the oscillations in the ignition circuit, all of the high-tension wiring, and the low-tension wiring associated with it, is enclosed in a continuous metal shield which makes good contact with the engine. How to secure a structure that will shield the high-tension circuits while providing good insulation under severe conditions of vibration, temperature, altitude, and exposure, is the problem.

In addition to being a secure structure providing mechanical protection to the wiring, the shield must fulfill exacting electrical requirements. All parts of the structure must be maintained at ground potential. Proper "grounding" means very low resistance joints and connections and grounding at frequent intervals. The use of ungrounded lengths of conduit will result in radiation from circulating high-frequency currents sufficient to cause a prohibitive amount of radio noise. Connecting tubes or conduit of high specific resistance will also cause difficulty.

What is a high-resistance ground or connection in the shield? To answer this question radio interference measurements must be made. In test work using calibrated equipment as specified in Navy Aeronautical Specification R-18a, it was found that some connections are more critical in this respect than others, probably because of the configuration of the shield and the resulting development of high-frequency charges and circulating currents. Interference becomes marked if the resistance across a joint or at a grounding point is over 0.002 ohms. At 0.004-0.010 ohms, interference is serious. Research shows that a safe resistance to be specified for connections in a new shield should not exceed 0.001 ohms. To obtain low-resistance joints machined bearing surfaces of adequate area must be provided. Non-metallic gaskets or anodized surfaces are a source of trouble and should be eliminated from the design.

A further requirement is that grounding points should be not more than 18 in. apart. This is found to be an essential requirement, modified at certain locations by the geometry of the shielding conduit. It is sometimes found that complete elimination of one grounding point will do no harm, while another ground close by will cause trouble if it is removed. With radio reception using high-sensitivity receivers operating on wave lengths as low as 1 or 2 m, bonding or grounding is preferably done at more frequent intervals, not more than 9 in. apart. The use of shields having a specific resistance of more than 0.005 ohms per ft also requires grounding at more frequent intervals.



■ Fig. 1 – Unshielded ignition radiation – vertical probe antenna 90 ft from ignition system

[This paper was presented at the National Aeronautic Meeting of the Society, New York, N. Y., March 12, 1942.]

[1] See "Aircraft Electricity," by Norman J. Clark and Howard E. Corbitt, Ronald Press, New York, N. Y., 1941.

[2] See SAE Transactions, 1941, pp. 107-116: "Supercharged Aircraft Ignition Harnesses," by Carl E. Swanson; also discussion by E. K. Von Mertens.

[3] See "Dielectric Phenomena in High-Voltage Engineering," by F. W. Peek, Jr., McGraw-Hill Book Co., New York, N. Y., 1928.

[4] See ASTM Standard D-495-41.

RADIO SHIELDING

by D. W. RANDOLPH

formerly Apex Electrical Mfg. Co.; now Lieutenant-Commander, U.S.N.R., Bureau of Aeronautics

THIS paper outlines the important electrical requirements of radio ignition shielding. The reason for low resistance grounding and bonding of the shield is explained.

A method for testing the resistance of plastic insulators to flash-over is suggested, and a method developed in England for reducing spark-plug electrode erosion is described.

★ ★ ★

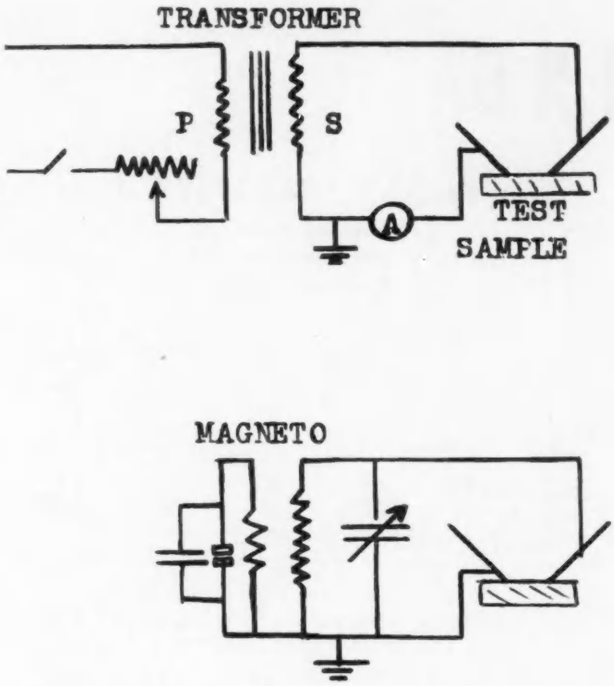
THE AUTHOR: D. W. RANDOLPH (M '23) received his B. S. degree in mechanical engineering from the Massachusetts Institute of Technology in 1921. He joined the Naval Reserve Flying Corps, and later worked for the Wright-Martin Aircraft Corp., the Bureau of Standards, the AC Spark Plug Division of GM, and the Aircraft Engine Division of the Packard Motor Car Co. In October, 1942, Mr. Randolph left his position as chief engineer of the Apex Electrical Mfg. Co., Cleveland, to become Lieutenant-Commander, Bureau of Aeronautics, Navy Department.

Measurements of interference in a complete engine installation are slow and time-consuming. It is not necessary that the inspection of each engine installation be made by radio interference measurements. Proper design and location of grounding points, bonds, and so on, having been demonstrated, inspection requirements can be met by the measurement of joint resistances. These measurements are made easily and quickly by means of a self-contained instrument operating on the principle of the Kelvin bridge. Resistance measurements to make sure of correct bonding of aircraft parts are carried out in the same manner, and the low resistances required of connections for bonding are comparable with those found necessary in the radio ignition shields.[1]

An excellent discussion has been written by C. E. Swanson of the reasons for difficulties encountered in maintaining perfect electrical insulation inside the shield.[2] Moisture, corona, and acid products of the decomposition of the air in the harness tubes vie with one another and with the reduction in the density of the air at higher altitudes in overstressing and destroying the compact insulation inside the shields. A remedy is to maintain a pressure inside the harness sufficient to prevent the entrance of moisture or other liquids and, at the same time, to allow sufficient air leakage from the shield to carry off the products of corona or spark-plug leakage before they have time to cause serious damage. The drawback to this system is the necessity of supplying a source of air under pressure which must be dried and piped to the harness connections. The added weight, complication, and servicing required are justified in airline service, but are hard to provide in a military engine where space and weight are of more vital importance. Another solution that seems equally successful is the provision of a filling or sealing compound in which the individual conductors are embedded and by means of which air and moisture are excluded. At the spark-plug connectors there is still an insulator and a small compartment that cannot be fully sealed unless the leakage through the spark plug can be eliminated completely. Leakage has been reduced to a remarkable degree in the newer ceramic spark plugs, but there is always the possibility that some will take place and the best solution for this difficulty lies in the provision of a small opening through which the gases may escape. Insulation inside the spark plug well must be given careful study.

A frequently neglected factor in the design of insulation for high-voltage circuits is the effect of material and dimensions on the electrostatic fields surrounding the insulator. The voltage stresses in the air surrounding an insulator may be very large and can cause frequent failures even though the materials used may be electrically and mechanically perfect. A study should be made of all air-surrounded cables and terminals to determine that materials and dimensions are suitable. The methods developed by Peek for calculating electrostatic stresses are simple to use and give results in full agreement with tests.[3]

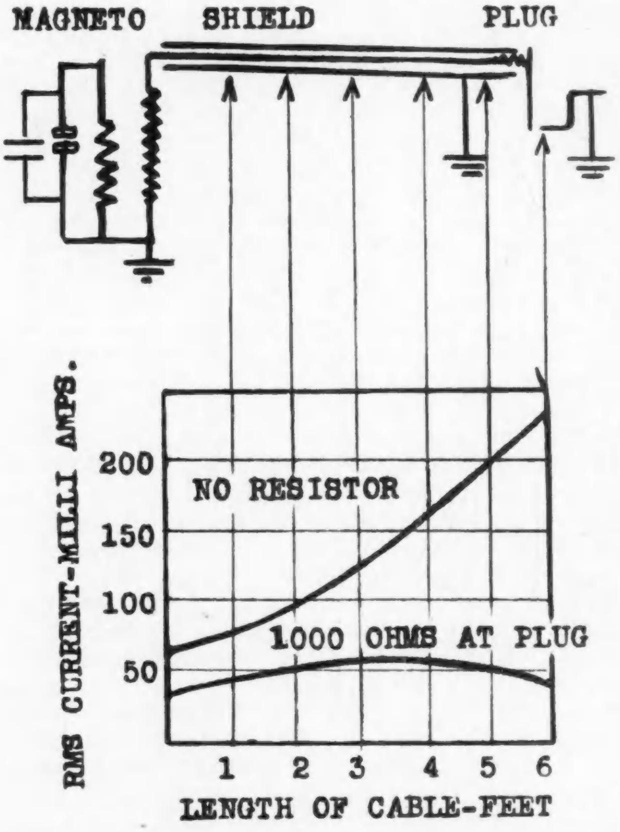
Insulation for ignition circuits must maintain high surface resistance under adverse conditions and, in addition, should be practically immune to "tracking," or the formation of conducting surface paths. Ceramic insulators in general are most capable of meeting this requirement but may not always prove easy of adaptation in a particular design. An interesting method for evaluating the resistance of a material to surface tracking has been developed and standardized by the American Society for Testing Materials.[4] In this test a pair of pointed tungsten electrodes are pressed against the insulator surface and a controlled alternating current arc is passed between the points. The time required to cause failure of the surface is noted. In making tests on plastic materials for high-voltage insulation in magneto circuits it was noticed that this test did not always predict failures that were observed in engine tests. To investigate this matter further the transformer used in the ASTM test was replaced by a magneto. The results obtained were not in agreement with the 60-cycle arc tests. As expected the time required for the formation of a permanent track on the insulator surface is a function of the speed of the magneto, that is of the number of sparks applied per minute. In addition the effect of added capacitance in the circuit is to increase the apparent resistance of the surface to tracking, so that a correct evaluation of the



■ Fig. 2 – ASTM arc resistance test as modified by use of magneto

utility of a given material must be made in terms of its location in the ignition circuit. The closer the insulator is to the magneto, the more drastic is the action of the spark on surface flash-over. Thus the plastic parts inside the magneto receive the most severe duty from this standpoint – those at the spark plug the least. The use of various coatings on the insulator surface to reduce the tendency to track has been suggested. Sixty-cycle tests of plastics coated with urea-formaldehyde varnishes gave very encouraging results, but tests repeated with the magneto as a source of voltage did not give the same satisfactory results. Very rapid failure of such coatings was observed, even where extra thick coatings were applied by means of successive layers of the varnish. In some cases the failure of the surface was not delayed at all by the addition of the coating. Exposure of the coated surface to temperature changes and to moisture or the products of air decomposition by an electric spark caused very rapid decreases in the ability of most coated surfaces to resist arcing from the magneto spark. The best all-around results were obtained from plastic insulators composed of melamine compounds and from ceramic insulators. The formation of even a faint track on an insulator surface is a frequent cause of ignition failure and a source of noise in the radio receiver. Hard rubbers of various compositions and sources were found to be widely different in their resistance to tracking, and no really successful coating for such a surface was obtained. Fig. 2 shows the original ASTM test circuit and the modified magneto test.

As the cable conductors pass through the shield to the spark plug, they are surrounded completely by the grounded shield. Because the result is a large increase in capacity of each lead, a low-capacity cable with stainless steel or other high-tensile cores of small diameter is used. The effect of capacity on the voltage output of the spark generator is well known and it acts as an electrical load that must be carried by the magneto. A capacity in excess of 200 mmf is excessive although certain long leads in modern shields exceed this value. Not only does an increase in the capacity of the lead throw an unwelcome additional responsibility on the magneto, but it also causes additional wear on the spark-plug electrodes. A method of reducing this wear is used in the KLG spark-plug connector on the Rolls-Royce engine. This method involves the use of a small non-inductive resistor in the high-tension lead as close as possible to the spark plug. The resistor has no effect on the radio shielding properties of the harness assembly and is not to be confused with the high resistances used in motor-car ignition circuits to minimize radio interference. The resistors used in aircraft magneto circuits have a resistance of from several hundred to a thousand ohms. They must be applied at a point near the spark plug as they depend for their effect on a reduction in the initial heavy current discharge across the plug gap which results from the sudden discharge of the capacity of the cable between the magneto and the plug. Fig. 3 shows a diagram of the electric circuits involved. The reduction in electrode wear is worth while, particularly if the resistor can be supplied as a built-in part of the spark plug instead of as a separate unit. A detailed description of the action of these resistors in reducing spark plug gap wear is given by G. E. Bairsto.[5] Briefly, the resistance adds damping to the circuit, so that the destructive effects of high average currents in the spark discharge are reduced. Measurements



■ Fig. 3 – Average current in shielded ignition circuit as modified by damping resistor

[5] See "Some Factors Controlling the Development of Electrical Ignition of Aero Engines," by G. E. Bairsto, lecture presented before the Royal Aeronautical Society, Feb. 16, 1939.

of the current in the spark-plug circuit show the effect of such resistors quite clearly and also demonstrate the fact that the damping resistance must be applied at the spark-plug end of the shielded lead. Measurements made to check those reported by Bairsto have been made by inserting a small thermocouple type of hot-wire ammeter in the return shield at various points in a lead from the magneto to the spark plug. Typical results are shown in Fig. 3. The reduction in electrode wear is approximately proportional to the reduction in the root-mean-square current. Tests reported by Bairsto, confirmed by others made in England and by tests made in this country, show that, with shielded leads having a capacitance of 200 mmf, equivalent to 6 ft of shielded ignition wire, the reduction in current and in gap wear is marked. Fig. 4 shows results of such a test and demonstrates the fact that more than 1000 ohms are not necessary.

Very confusing and contradictory results have been obtained in engine tests by the use of resistors that do not maintain their resistance under the conditions encountered in the ignition system. The effect of current flow at temperatures of 300-400 F produces a permanent change in the resistance of some resistor materials. A change in resistance of several times the original value often takes place. Some semi-conductors used for small resistors have non-linear volt-ampere characteristics – that is, their resistance depends on the voltage applied to them. A resistance of 500 ohms at 6 v may become only 50 ohms if the voltage across it is 500. Such a resistor is useful as a lightning arrestor but will not reduce the current in the ignition circuit. The resistor selected should not show more than a few per cent change in resistance if the measuring voltage is varied by a factor of 100. Certain carborundum compounds have been found satisfactory for this use. The temperature coefficient of resistance must not be so great as to cause a change in resistance of more than 10% at temperatures up to 400 F. In one engine test resistors of a nominal 800 ohms were used. During the test it was observed that there was no decrease in plug electrode wear after 40 hr of engine run. The resistors were checked and found to have changed physically so that their resistance had decreased to less than 50 ohms.

The wire in the usual unfilled harness is subjected to great electrical stress and must maintain its insulation in spite of abrasion, high temperature, and the effects of corona on the rubber or other insulating compounds used. Because many harnesses are wired by pulling individual cables through the conduits, the cable must be able to withstand considerable tensile pull. This requirement puts an additional obstacle in the path of the cable manufacturer in attempting to obtain the ideal cable insulation. A recently added requirement of considerable practical importance is the provision of a waterproofing treatment to the cotton braid used to protect the cable from abrasion. Many cable failures were caused by the penetration of moisture into the braid; the high voltage then followed the cotton semi-conductor until a weak spot in the inner core insulation gave way and a puncture was the result. The use of small-diameter stainless-steel wire to provide strength and minimum capacity is a distinct improvement, as is the production of plastic insulating material with excellent anti-tracking qualities.

Effective testing of the assembled harness to detect electrical leaks can best be done by means of high-voltage direct current, with a suitable means to limit the amount of current through a fault and the provision of an indicator to disclose the intermittent current flow due to insulator flash-over. The use of alternating current from a transformer test set may be justified for rapid testing of new shields, but has the disadvantage that, when an insulation failure is disclosed, considerable damage is usually done before the current can be shut off. In testing the shields on an engine, the use of controlled and measured direct current will locate a fault just as efficiently and will, at the

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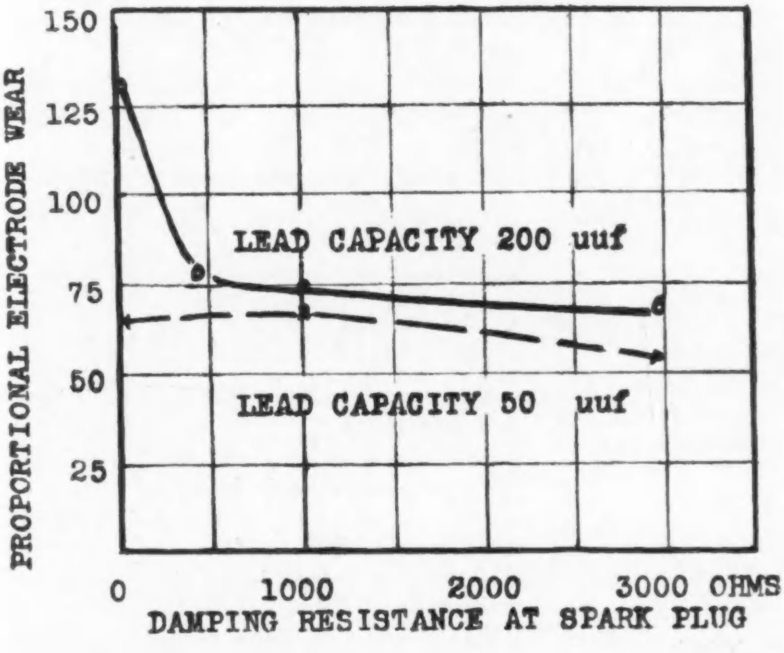


■ Fig. 4 – Spark-plug electrode wear as affected by the addition of damping resistance

Design of CROSS-FLOW

by **PAUL A. SCHERER**
AiResearch Mfg. Co.

THIS paper presents a simple and rapid method of determining the performance of cross-flow intercoolers, oil coolers, or Prestone radiators from laboratory tests of a model or basic unit of the cooler. The method lends itself equally as well to the determination of the size of a cooler of any set performance.

Due to the comparative rapidity with which these calculations can be made, it becomes, with the use of this method, an easy matter to make a series of calculations to determine the relations between lengths, cooling air flow, and pressure drops, for any desired performance.

■ ■ ■

THE AUTHOR: PAUL A. SCHERER, research director of the AiResearch Mfg. Co., joined this company in 1940. Prior to that time he had been a consultant-engineer in refrigeration and air conditioning in San Francisco. He was associated with the design of the air conditioning installation for the Dallas plant of North American Aviation, Inc.

INCREASINGLY severe conditions of aircraft operation have made it difficult for powerplant engineers to estimate the size and weight of oil coolers, intercoolers, and Prestone radiators for required performance.

While a great deal of work is now being done at the NACA and elsewhere, including our own company, on the calculation of various core areas, this paper is limited to extending the performance of laboratory-tested core sections to actual problems of flight. It is therefore assumed that laboratory tests have been made on the type of cores which it is desired to study. Such laboratory tests would show that a unit of certain definite dimensions has definite heat-dissipation values for particular weight-flow rates of fluid streams which are rejecting and absorbing heat.

■ Symbols

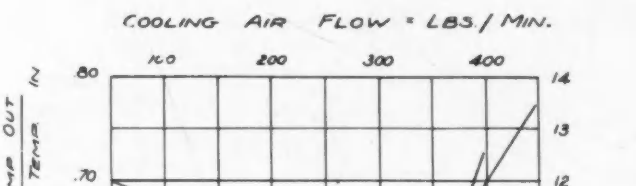
The definition of the following symbols has been written to apply to a supercharger intercooler. It is desired to point out that, for general use, the word "air" should be superseded by "fluid," and the terms "supercharger," "carburetor," or "engine" apply to the fluid stream rejecting heat, while the term "coolant" applies to the fluid stream absorbing heat.

Ac	cross-sectional area of coolant air stream in the design intercooler – sq ft.
AcB	cross-sectional area of coolant air stream in the basic intercooler – sq ft.
As	cross-sectional area of supercharger air stream in the design intercooler – sq ft.
AsB	cross-sectional area of supercharger air stream in the basic intercooler – sq ft.
B	indicates basic intercooler – laboratory tested model.
Bar.	barometer – in. hg.
c	indicates coolant air stream.
Cm, Cp	mean specific heat – Btu per lb per deg F.
E	effectiveness of the intercooler – the temperature drop of supercharger air stream divided by initial temperature difference between the supercharger and coolant air streams – $\dfrac{(To - Tf)}{(To - t_0)}$.
G	unit area weight flow – lb per cross-sectional sq ft min.
IAS	indicated air speed – in. H_2O.
Lc	coolant fluid stream length of the design intercooler – in.
LcB	coolant fluid stream length of the basic intercooler – in.
Ln	no flow height of the design intercooler – in.
LnB	no flow height of the basic intercooler – in.
Ls	supercharger fluid stream length of the design intercooler – in.
LsB	supercharger fluid stream length of the basic intercooler – in.
Δpc	coolant air static pressure drop in the design intercooler – in. H_2O.
ΔpcB	coolant air static pressure drop in the basic intercooler – in. H_2O.
Δps	supercharger air static pressure drop in the design intercooler – in. H_2O or in. hg.
ΔpsB	supercharger air static pressure drop in the basic intercooler – in. H_2O or in. hg.
R	ratio of coolant to supercharger air weight flow – dimensionless.
Ra	rating number of the design intercooler – dimensionless.

[This paper was presented at a meeting of the Southern California Section of the Society, Los Angeles, Calif., March 13, 1942.]

HEAT EXCHANGERS from Tested Core Sections

RaB	rating number of the basic intercooler – dimensionless.
S	surface – sq ft.
s	indicates carburetor, engine, or supercharger air stream.
σ	density ratio of actual air to standard air (59 F – 29.92 Bar.) – dimensionless.
To	temperature of supercharger air to intercooler – deg F.
Tf	temperature of supercharger air from intercooler – deg F.
t_0	temperature of coolant air to intercooler – deg F.
tf	temperature of coolant air from intercooler – deg F.
U	overall heat-transfer coefficient – Btu per hr per sq ft per deg F.
Wc	coolant air weight flow in the design intercooler – lb per min.
WcB	coolant air weight flow in the basic intercooler – lb per min.
Ws	supercharger air weight flow in the design intercooler – lb per min.
WsB	supercharger air weight flow in the basic intercooler – lb per min.



■ **Fig. 1 – Typical performance of a laboratory-tested intercooler**

■ Laboratory Test Data

Fig. 1 shows a test curve and illustrates one method of presenting laboratory test data. This particular curve is for heat transfer from one air stream to another with the two streams moving at right angles. It therefore shows the performance of a typical supercharger intercooler.

The horizontal scale at the base of the curve sheet indicates the flow in pounds per minute of air from the supercharger (Ws). The horizontal scale at the top of the sheet shows the weight flow for the coolant air stream (Wc). Associated with each of these two scales is a corresponding curve sloping upwards on the right and designated Supercharger Air-$\sigma\Delta p$ and Coolant Air-$\sigma\Delta p$. The static pressure drop in inches of water of standard air required to produce this weight flow may be read directly on the extreme right-hand vertical scale. Standard air, as here used, is air at 59 F and a barometer (absolute pressure) of 29.92 in. hg.

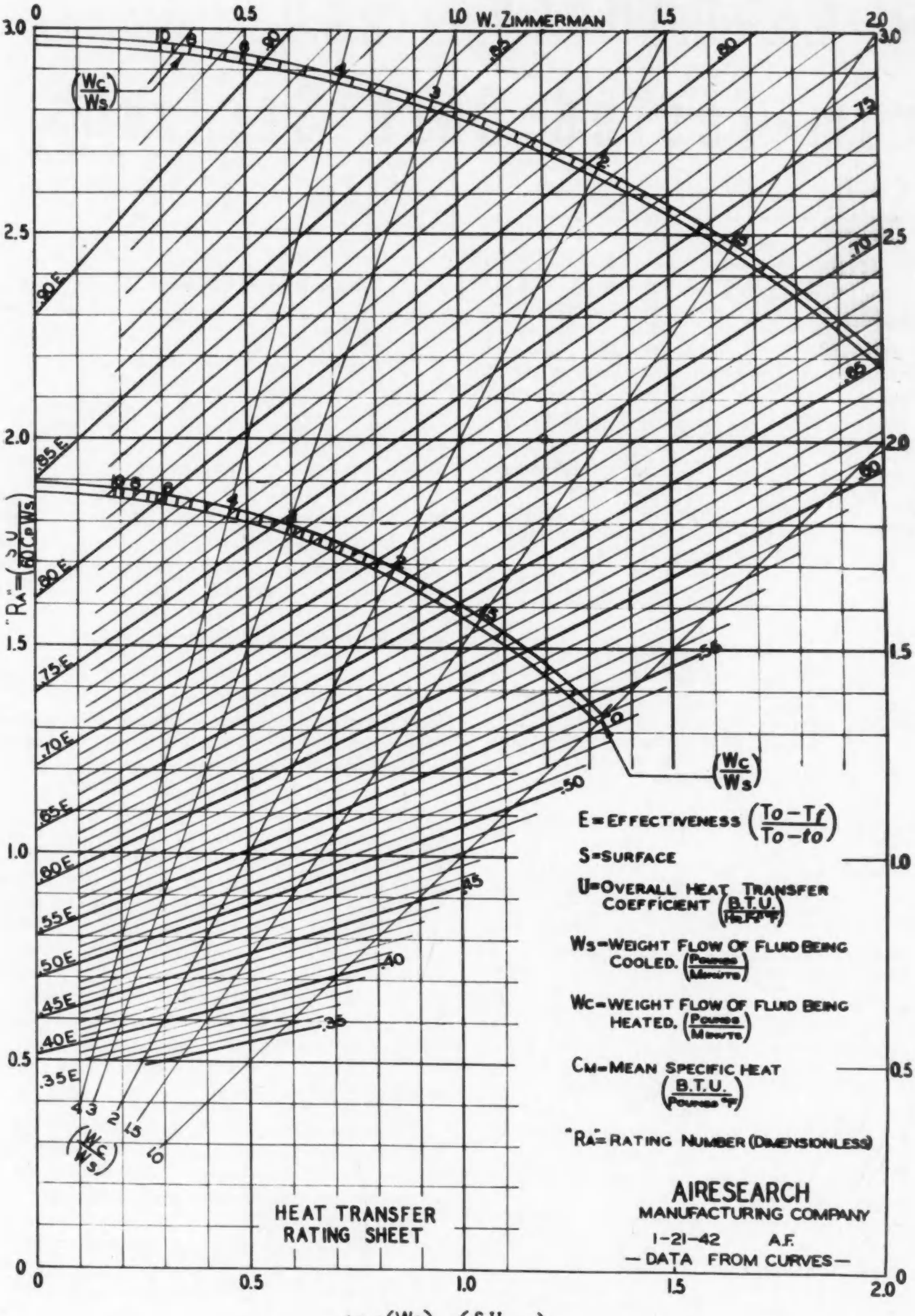
■ Illustrative Problem

Let us assume that we have a particular problem to solve. Although the technique applies to any heat-transfer unit in cross flow, a supercharger intercooler will be selected, since air-to-air heat transfer more graphically illustrates changes in final temperature, pressure drops, and dimensions. The design and duty of the airplane enable us to set up the intercooler problem.

a. The point of critical performance is estimated.

b. The required weight of air flow (Ws) is obtained from the engine manufacturer.

c. The allowable pressure drop of supercharger air (ΔPs) through the intercooler is determined. In setting the pressure drop it is desirable to select a range between the maximum and the minimum values allowable so that



■ Fig. 2 – Heat-transfer rating sheet for cross flow

the final choice may be made with reference to other variables affecting the efficiency of the intercooler.

d. The indicated air speed (IAS) for the critical problem is known.

e. The entrance, exit, and duct losses are estimated. Thus the coolant air stream static pressure drop across the intercooler (Δpc) can be determined. This value will be the maximum available static pressure drop. Since intercooler drag is nearly directly related to the weight flow of the coolant air stream, it is well to select for study a range in pressure drops below the maximum allowable value.

f. The temperature of the supercharger air entering the intercooler (To) is estimated from the characteristics of the supercharger and the available ram pressure.

g. The temperature of the supercharger air leaving the intercooler (Tf) is fixed by the requirements of the engine manufacturer, if not by the contract. Thought should be given to the factors of intake manifold pressure and supercharger power input as related to temperature, and the effect on net engine power available for flight. An analysis of the density of the charge in the manifold and detonation characteristics will show that final temperature alone is a poor criterion of optimum performance.

h. Space limitations, at least as to certain dimensions, will exist.

Therefore, prior to seeking a solution of the problem, there are certain given requirements, which may include the following:

	Supercharger Stream	Coolant Stream
Weight Flow (lb per min)	Ws	Wc
Allowable static pressure drop (in. H_2O)	Δps	Δpc
Barometer (in. hg)	Bar.s	Bar.c
Temperature at Entrance (F)	To	t_0
Temperature at Exit (F)	Tf	tf
Length of Flow (in.)	Ls	Lc
Length of No Flow (in.)	Ln	

Method of Solution

The next steps in the procedure are the determination of quantities as described in the following:

1. The equivalent supercharger air flow for the basic intercooler (WsB) is that flow which has the same unit area weight flow (G) as the design unit. This is then the supercharger flow of the design intercooler multiplied by the ratio of the basic to design supercharger header areas.

$$WsB = Ws \times \frac{AsB}{As}$$

2. The design intercooler coolant air pressure drop ($\sigma\Delta pc$) in inches of water of standard air required to produce the weight flow of the problem is the product of the actual pressure drop (Δpc) and the density ratio (σ) of the mean coolant air to standard air.

3. The basic intercooler coolant air pressure drop corrected to standard air ($\sigma\Delta pcB$) is determined on a general assumption that the entrance and exit losses are 22% of the design intercooler pressure drop. After this deduction is made, the remaining friction loss is multiplied by the coolant air stream length ratio $\left(\frac{LcB}{Lc}\right)$. Then the entrance and exit losses are re-added.

$$\sigma\Delta pcB = 0.78\ (\sigma\Delta pc)\ \frac{LcB}{Lc} + 0.22\ (\sigma\Delta pc)$$

4. The effectiveness of the basic intercooler (EB) at the equivalent supercharger air weight flow (WsB) is read from Fig. 1 by entering the lower horizontal scale at the value from step 1 and moving upwards to the proper position in the family of curves sloping upwards to the left for the value of the basic intercooler coolant air pressure drop corrected to standard air ($\sigma\Delta pcB$) from step 3. The effectiveness is then read by moving horizontally to the left-hand scale.

5. The coolant air flow of the basic intercooler (WcB) is then read from Fig. 1 by entering the vertical scale on the extreme right at the value from step 3 of the basic intercooler coolant air pressure drop corrected to standard air ($\sigma\Delta pcB$), and moving horizontally to the left to the intersection with the Coolant Air-$\sigma\Delta p$ line. The coolant air weight flow is then read by moving vertically upwards to the top horizontal scale.

6. The basic unit coolant air to supercharger air weight flow ratio (RB) is obtained by dividing the basic intercooler coolant air weight flow (WcB) from step 5 by the basic intercooler supercharger air weight flow (WsB) from step 1.

$$RB = \frac{WcB}{WsB}$$

7. The basic intercooler rating number (RaB) for the condition of operation as determined in the previous six steps is read from Fig. 2. Fig. 2 is designated as a "heat transfer rating sheet for cross flow." The vertical scale is a plot of a dimensionless number called a "Rating Number" (Ra). While for this work it is not necessary to break down the structure of this number, its derivation from the product of a corrected active surface (S) and the overall heat transfer coefficient (U) divided by the "water equivalent" of the supercharger fluid flow, is shown. The horizontal scale is the same rating number divided by the weight flow ratio ($R = Wc/Ws$). As a convenience, the two arc scales and the origin permit the direct drawing of a line for any weight flow ratio.

This chart was prepared by W. Zimmerman of our company, from basic calculations carried to four significant figures obtained by using the method of calculation proposed by Nusselt[1]. The basic Newtonian assumption was made that, in heat transfer from a fluid to a body, the rate of change of temperature difference is proportional to the temperature difference itself. Three equations were then written for increments of heat transferred in unit time. One of these was in terms of temperature difference, surface, and the heat-transfer coefficient. The other two were in terms of heat rejected by the supercharger stream, and of heat absorbed by the coolant stream. For the integration for right-angle cross flow after the method of Volterra, an assumption of uniform velocity distribution was made. It should be here noted that the velocity distribution in cross-flow heat exchangers can only be uniform when no heat is being transferred.

This much detail has been included here to indicate the usefulness of the chart for any two fluids in right-angle

[1] See *Techn. Mechan. u. Thermodynamik*, Bd. 1, Nr. 12, December, 1930, pp. 417-422: *"Eine neue Formel für den Wärmedurchgang im Kreuzstrom,"* by Wilhelm Nusselt.

cross flow. Should this derivation be desired, our company will be pleased to furnish the paper upon request.

The rating number for the basic intercooler can then be found by moving from the origin up the weight flow ratio line of the value of $\left(RB = \frac{WcB}{WsB}\right)$ from step 6 to its intersection with the effectiveness line of the value of the basic intercooler effectiveness (EB) from step 4. The rating number (RaB) is then read by moving horizontally to the right or left to the rating number scale.

8. The weight flow of coolant air in the design intercooler (Wc) is the product of the weight flow of coolant air in the basic intercooler (WcB) and the area ratio of the design to basic coolant air headers.

$$Wc = WcB \times \frac{Ac}{AcB}$$

9. The design intercooler coolant air to supercharger air weight flow ratio (R) is obtained by dividing the design intercooler coolant air weight flow (Wc) from step 8 by the design intercooler supercharger air weight flow (Ws) from part "b" of the problem statement.

$$R = \frac{Wc}{Ws}$$

10. The rating number of the design intercooler (Ra) is obtained from the rating number of the basic intercooler. As the value of the unit area weight flow of the supercharger air is the same for both intercoolers, the rating numbers will be proportional to the supercharger air stream lengths. The design intercooler rating number (Ra) will then be the product of the basic intercooler rating number (RaB) from step 7 and the design to basic intercooler supercharger air stream length ratio $\left(\frac{Ls}{LsB}\right)$.

$$Ra = RaB \times \frac{Ls}{LsB}$$

11. The effectiveness of the design intercooler (E) is read from Fig. 2 by moving up the line of the design intercooler coolant air to supercharger air weight flow ratio

$$\left(R = \frac{Wc}{Ws}\right)$$

value from step 9 to the intersection with the design intercooler rating number (Ra). The effectiveness is then read by following the effectiveness curves to their scale on the left.

12. The drop in supercharger air temperature ($-\Delta T$) is the product of the effectiveness (E) from step 11 and the difference between the supercharger and coolant air initial temperatures (To-to) from the statement of problem requirements.

$$-\Delta T = (E) \times (To - t_0)$$

The final supercharger air temperature (Tf) then follows. $Tf = To - (-\Delta T)$

13. The rise in coolant air temperature (Δt) is equal to the supercharger air temperature change ($-\Delta T$) from step 12 divided by the ratio of coolant to supercharger air weight flows $\left(\frac{Wc}{Ws} = R\right)$ from step 9.

$$\Delta t = -\frac{\Delta T}{R}$$

The final coolant air temperature (tf) then follows: $tf = to + \Delta t$.

14. The basic intercooler supercharger air pressure drop corrected to standard air ($\sigma\Delta psB$) is read from Fig. 1 by entering the lower horizontal scale at the basic intercooler supercharger air weight flow (WsB) value from step 1 and moving upwards to the Supercharger Air-$\sigma\Delta p$ line. The value of the pressure drop is then read by moving horizontally to the scale on the extreme right.

15. The design intercooler supercharger air pressure drop corrected to standard air ($\sigma\Delta Ps$) is determined from the basic intercooler supercharger air pressure drop corrected to standard air ($\sigma\Delta psB$) obtained in step 14.

Again a fair general assumption of entrance and exit losses would be 22%. After this deduction is made, the remaining friction loss is multiplied by the length ratio of the design to basic intercooler supercharger air stream length $\left(\frac{Ls}{LsB}\right)$. Then the entrance and exit losses are re-added.

$$\sigma\Delta ps = 0.78\ (\sigma\Delta psB)\ \frac{Ls}{LsB} + 0.22\ (\sigma\Delta psB).$$

16. The actual static pressure drop at the problem (Δps) is obtained by dividing the static pressure drop corrected to standard air ($\sigma\Delta ps$) from step 15 by the density ratio (σ) of the supercharger air at the problem to standard air.

$$\Delta ps = \frac{\sigma\Delta ps}{\sigma}$$

In this connection, it is well to state that the pressure (Bar.) and absolute temperature ($T+460$) for determining the density ratio should be taken at the mean of entrance and exit conditions. The arithmetical mean is sufficiently accurate. However, a geometric mean is close to the logarithmic mean and not difficult to obtain on a slide rule.

Fig. 3 is a numerical example problem for illustrative purposes. The equation numbers correspond to the step numbers as just discussed.

The method is now explained. Obviously, the process can be used in any direction. Often it might be desirable from a given effectiveness and mass ratio immediately to determine the "rating number" required. If laboratory test data have been reduced to performance of cores of common dimensions, the examination of the data to find the core or cores most nearly satisfying a design condition is made fairly simple by reducing the design intercooler rating number (Ra) and coolant air to supercharger air weight flow ratio (R) to those of a core of the same common dimensions.

With one unit designed to approximate the problem conditions, certain dimensions can be altered and either accurate or approximate estimates run directly. Changes in stream lengths will change the "rating numbers" and weight flow ratios in direct proportion if the unit area mass flows (G) are held constant. Changing the supercharger length causes a vertical translation of position on the Rating Sheet for Cross Flow (Fig. 2). Changing the coolant length causes a horizontal translation of position. Changing the no-flow length causes no change in position on the Rating Sheet. Then a combination change of the last two changes is in a horizontal translation. This means that the supercharger air weight flow can be held constant by changing the no-flow length in inverse proportion to the cooling air length change ratio, and the translation of position on the Rating Sheet will be horizontal in proportion to the cooling air length ratio. It should be noted here

that, for a constant unit area mass flow, the pressure drop varies with the length, as discussed in step 3.

In cases where the unit area mass flows are not held constant, a change in the coolant stream length, without a change in the supercharger header area and static pressure drop, will change the film coefficient roughly inversely with the square root of the length ratio[2]. Multiplying the "rating number" by one half of this value (expressed in percentage) will give an approximate new "rating number" and an approximate answer when used with the similarly modified weight flow ratio. The same reasoning can be applied to changes in the supercharger header dimensions.

[2] See the *Journal of the Aeronautical Sciences*, Vol. 8, No. 7, May, 1941, pp. 292-299: "Method of Computing the Dimensions of Airplane Engine Intercoolers," by J. Kendall Thornton and Joseph G. Beerer.

Naturally any such modifications should be checked by a re-run in the usual manner. By making several calculations, data can be obtained for various curves, giving the relations between lengths and coolant air flow for specified conditions, such as calculated by Thornton and Beerer[2], as well as any other relations that might be desired.

Other Design Considerations

There are certain points to be watched with some care:

I. An allowance must be made for suitable flanges in obtaining core dimensions (which apply only to the volume swept by both air streams).

II. The ducting for the design intercooler installation may give a different velocity distribution at the face of the intercooler from that of the laboratory core tests.

Given:

Dimensions: Ls = 26" Lc = 8" Ln = 16"

Supercharger		Coolant	
Ws =	90 lbs/min		
To =	300° F	to =	+10 ΔR = 290
		Δp =	15" H_2O
Bar. (at face) =	31" Hg	Bar. =	11.10" Hg (25,000 ft. alt.)

Find:

Performance using core shown Fig. 1

Dimensions: LsB = 20.75" LcB = 10.25" LnB = 13"

- -

1) $WsB = 90 \times \frac{10.25 \times 13}{8 \times 16} = 93.7$ lbs/min.

2) $\sigma\Delta pc = 15 \times \frac{11.65 \times 519}{29.92 \times 515} = 5.9$ "H_2O

3) $\sigma\Delta pcB = 5.9 \times \frac{10.25}{8} = 7.58"\ H_2O$

4) W_sB & $\sigma\Delta pcB$ (Fig. 1) $\longrightarrow$ EB = .625

5) $\sigma\Delta pcB \xrightarrow{(Fig.1)} WcB = 305$ lbs/min.

6) $\frac{WcB}{WsB} = \frac{305}{93.7} = 3.22$

7) E_B & $\frac{WcB}{WsB}$ (Fig.2) $\longrightarrow R_aB = 1.16$

8) $Wc = 305 \times \frac{26 \times 16}{20.75 \times 13} = 4.71$ lbs/min.

9) $\frac{Wc}{Ws} = \frac{471}{90} = 5.23$

10) $Ra = 1.16 \times \frac{26}{20.75} = 1.45$

11) Ra & $\frac{Wc}{Ws}$ (Fig.2) $\longrightarrow$ E = .72

12) $-\Delta T = .72 \times 290 = 209^\circ$ F

13) $\Delta t = \frac{209}{5.23} = 40^\circ$ F

14) $\sigma\Delta psB$ (Fig.1) = 4.08 "H_2O

15) $\sigma\Delta ps < 4.08 \times \frac{26}{20.75} = 5.1"H_2O$

16) $\Delta ps < 5.1 \times \frac{29.92 \times 656}{30.5 \quad 519} = 6.3$ "H_2O = .5" Hg

- -

STATEMENT OF PERFORMANCE

Supercharger		Coolant	
To	= 300° F	to	= +10° F ΔR = 209
-ΔT	= 209° F	Δt	= 40° F E = .72
Tf	= 91° F	tf	= 50° F
Ws	= 90 lbs/min	Wc	= 471
Bars.	= 31" Hg	Bar.	= 11.10" Hg
Δps	= 6.3" H_2O = .5" Hg	Δpc	= 15" H_2O

Fig. 3 – Illustrative problem

III. The laboratory tests are usually run at a supercharger air inlet temperature obtained with a few pounds of steam pressure on the supercharger stream heating coil and normal room temperatures of the entering coolant stream. The pressure drop ratio will vary approximately with the 1/3 power of the absolute viscosity ratio. This probable thermal deviation from the critical flight conditions affects flow characteristics through modification of the Reynolds number. On preliminary design this deviation can be neglected. If these thermal corrections are applied to the laboratory test data, the calculated results show remarkable accuracy.

IV. A margin should be allowed for the consideration in sections II and III of, say, 5 F on the final supercharger air stream temperature (Tf) and about 1 in. H_2O pressure drop on each of the fluid streams for intercooler design.

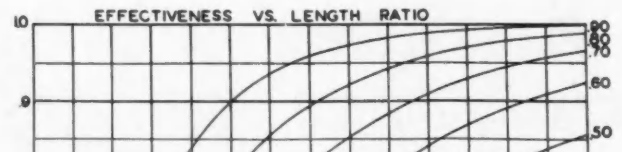
V. In the determination of the pressure drops for fluids with marked viscosity change with respect to temperature change, care must be exercised in the selection of suitable laboratory data.

Alternate Method

A family of curves (Fig. 4) is of great interest.

This chart was prepared by our S. K. Andersen and shows the change in effectiveness versus a particular stream length ratio when unit area weight flows are held constant. Mr. Andersen's explanation of the use of this chart is available on request. It can be used successively on each stream, bearing in mind that the ratio of effectiveness values of the two streams is equal to the reciprocal of the weight flow ratio. Then by the use of an imaginary intercooler as an intermediate step, the performance of the design intercooler can be obtained from the laboratory test data. It is not necessary, therefore, to use Nusselt factors[3] with either chart. The use of the two charts together gives a rather comprehensive picture of the changes occurring in an intercooler responsive to alteration of dimensions and weight flows. The two methods yield sufficiently close solutions in the field generally used for preliminary design.

[3] See p. 149: "Heat Transmission," by William H. McAdams, McGraw-Hill Book Co., New York, N. Y., 1933.



■ Fig. 4 – Effectiveness versus length ratio

Conclusion

This paper is not intended as more than an engineering aid. Nothing is contained which does not directly follow from published material. This course has been followed to avoid any questions as to the propriety of an unrestricted presentation.

Aircraft-Engine Radio Shielding

(*Concluded from page 541*)

same time, show whether it is an actual complete insulation failure or only a surface leakage condition that can be corrected by cleaning or drying. The use of such a testing device will disclose leakage across insulator surfaces due to changes in the surface after exposure to moisture. It also will show leakage caused by tracking due to occasional flash-over that may later on become serious enough to cause complete failure of one or more leads. All of this valuable information is lost when the usual alternating current test set is used. The megger will also prove of value in studying the cause of failure in harness insulation, and its use is so common as to need no comment. Electrical leakage paths that can be detected readily and their magnitude measured with a well-designed direct-current test set will often produce radio noises that can be detected by the more complicated radio interference-measuring equipment.

Mechanical problems having to do with support, effects of vibration, and chafing of cables and insulators need no discussion. In spite of a great deal of work, all these troubles are still evident. The recent tendency toward the design of an ignition system as a complete unit in itself seems to be the most logical and successful solution to the problem. Shielding, insulation, the spark generator and the spark plugs must be designed to work together to obtain the maximum in efficiency and in trouble-free operation. Future progress seems to lie in the use of improved shielding having no internal air spaces, in the use of improved plastic or ceramic insulators, and in the development of new ignition systems in which low-voltage currents are carried to individual spark coils in or near the spark plugs to remove the necessity for high-voltage distribution. Rapid progress has been made in recent months toward these objectives.

Index to Transactions

Author Index

THE Index for the Transactions Section of the SAE Journal for 1942 is printed on the following pages. Page numbers for this Section are consecutive from January to December. The table below indicates the pages contained in each issue:

January	1-36	May	169-208	September	373-416
February	37-72	June	209-252	October	417-464
March	73-124	July	253-320	November	465-508
April	125-168	August	321-372	December	509-548

Following the Transactions Index there is printed the General Editorial Section Index.

(E) indicates excerpt published.

AUTHOR INDEX

(E) indicates excerpt published.

Discusser Index

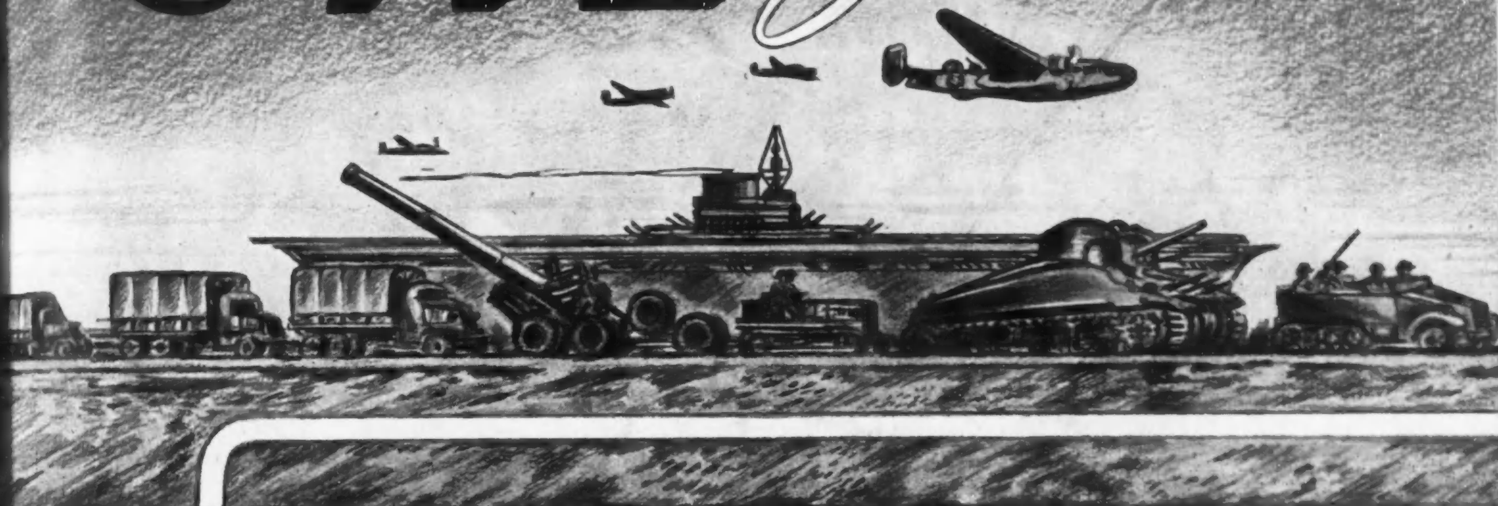
Subject Index

SUBJECT INDEX

SUBJECT INDEX

SUBJECT INDEX

S·A·E Journal

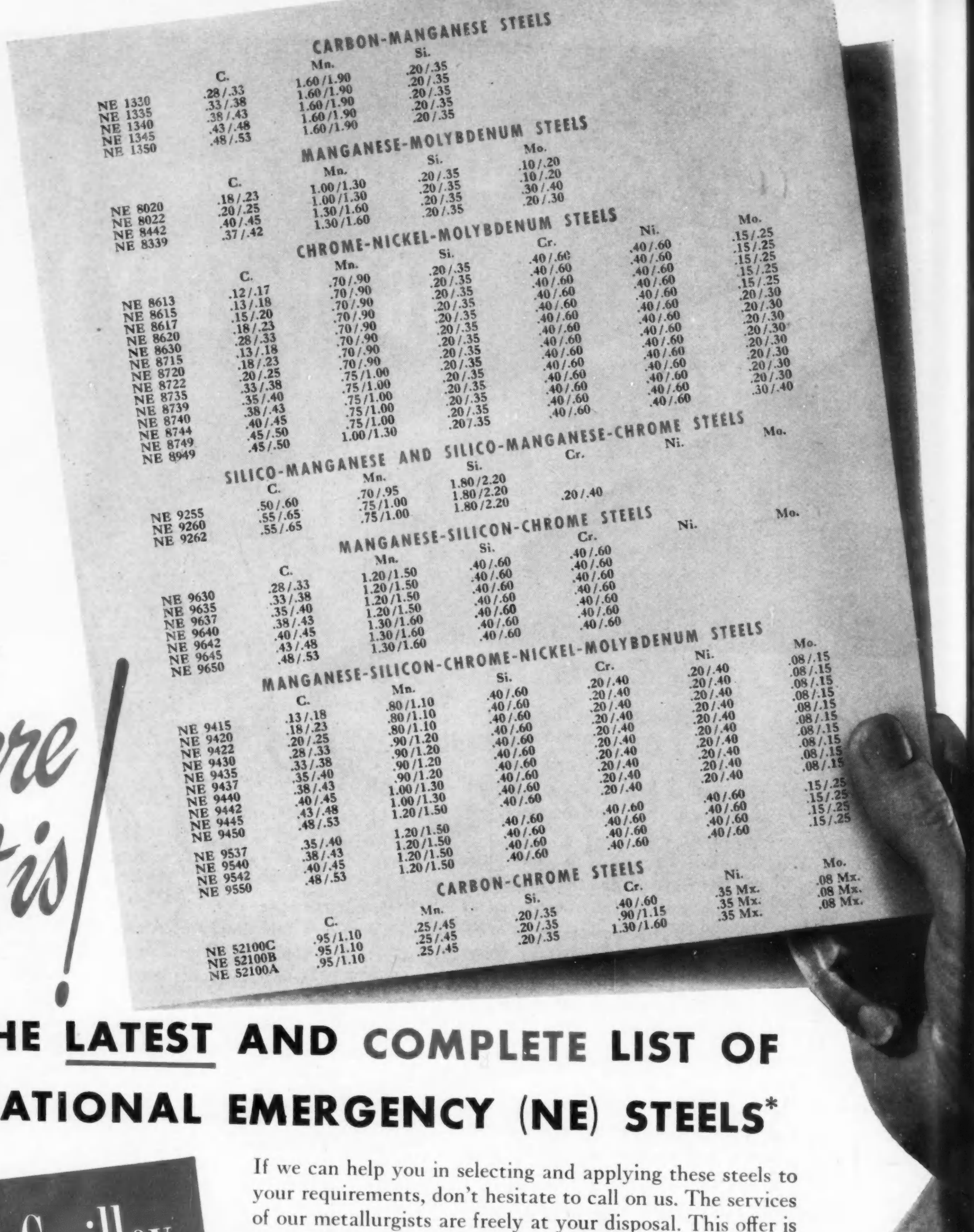


DECEMBER 1942

S·A·E

SOCIETY OF AUTOMOTIVE ENGINEERS

SAE JOURNAL, December, 1942, Vol. 50, No. 12. Published monthly by the Society of Automotive Engineers, Inc. Publication office at 56th and Chestnut Streets, Philadelphia, Pa. Editorial and advertising departments at the headquarters of the Society, 29 West 39th Street, New York, N. Y. $1 per number, $10 per year; foreign $12 per year; to members 50 cents per number, $5 per year. Entered as second class matter, Feb. 16, 1933, at the Post Office at Philadelphia, Pa., under the Act of Aug. 12, 1912. Acceptance for mailing at special rate of postage provided for in Section 1103 Act of Oct. 3, 1917, authorized on Jan. 14, 1928.



CARBON-MANGANESE STEELS

	C.	Mn.	Si.
NE 1330	.28/.33	1.60/1.90	.20/.35
NE 1335	.33/.38	1.60/1.90	.20/.35
NE 1340	.38/.43	1.60/1.90	.20/.35
NE 1345	.43/.48	1.60/1.90	.20/.35
NE 1350	.48/.53	1.60/1.90	.20/.35

MANGANESE-MOLYBDENUM STEELS

	C.	Mn.	Si.	Mo.
NE 8020	.18/.23	1.00/1.30	.20/.35	.10/.20
NE 8022	.20/.25	1.00/1.30	.20/.35	.10/.20
NE 8442	.40/.45	1.30/1.60	.20/.35	.30/.40
NE 8339	.37/.42	1.30/1.60	.20/.35	.20/.30

CHROME-NICKEL-MOLYBDENUM STEELS

	C.	Mn.	Si.	Cr.	Ni.	Mo.
NE 8613	.12/.17	.70/.90	.20/.35	.40/.60	.40/.60	.15/.25
NE 8615	.13/.18	.70/.90	.20/.35	.40/.60	.40/.60	.15/.25
NE 8617	.15/.20	.70/.90	.20/.35	.40/.60	.40/.60	.15/.25
NE 8620	.18/.23	.70/.90	.20/.35	.40/.60	.40/.60	.15/.25
NE 8630	.28/.33	.70/.90	.20/.35	.40/.60	.40/.60	.15/.25
NE 8715	.13/.18	.70/.90	.20/.35	.40/.60	.40/.60	.20/.30
NE 8720	.18/.23	.70/.90	.20/.35	.40/.60	.40/.60	.20/.30
NE 8722	.20/.25	.70/.90	.20/.35	.40/.60	.40/.60	.20/.30
NE 8735	.33/.38	.75/1.00	.20/.35	.40/.60	.40/.60	.20/.30
NE 8739	.35/.40	.75/1.00	.20/.35	.40/.60	.40/.60	.20/.30
NE 8740	.38/.43	.75/1.00	.20/.35	.40/.60	.40/.60	.20/.30
NE 8744	.40/.45	.75/1.00	.20/.35	.40/.60	.40/.60	.20/.30
NE 8749	.45/.50	.75/1.00	.20/.35	.40/.60	.40/.60	.20/.30
NE 8949	.45/.50	1.00/1.30	.20/.35	.40/.60	.40/.60	.30/.40

SILICO-MANGANESE AND SILICO-MANGANESE-CHROME STEELS

	C.	Mn.	Si.	Cr.	Ni.	Mo.
NE 9255	.50/.60	.70/.95	1.80/2.20			
NE 9260	.55/.65	.75/1.00	1.80/2.20			
NE 9262	.55/.65	.75/1.00	1.80/2.20	.20/.40		

MANGANESE-SILICON-CHROME STEELS

	C.	Mn.	Si.	Cr.	Ni.	Mo.
NE 9630	.28/.33	1.20/1.50	.40/.60	.40/.60		
NE 9635	.33/.38	1.20/1.50	.40/.60	.40/.60		
NE 9637	.35/.40	1.20/1.50	.40/.60	.40/.60		
NE 9640	.38/.43	1.20/1.50	.40/.60	.40/.60		
NE 9642	.40/.45	1.30/1.60	.40/.60	.40/.60		
NE 9645	.43/.48	1.30/1.60	.40/.60	.40/.60		
NE 9650	.48/.53	1.30/1.60	.40/.60	.40/.60		

MANGANESE-SILICON-CHROME-NICKEL-MOLYBDENUM STEELS

	C.	Mn.	Si.	Cr.	Ni.	Mo.
NE 9415	.13/.18	.80/1.10	.40/.60	.20/.40	.20/.40	.08/.15
NE 9420	.18/.23	.80/1.10	.40/.60	.20/.40	.20/.40	.08/.15
NE 9422	.20/.25	.80/1.10	.40/.60	.20/.40	.20/.40	.08/.15
NE 9430	.28/.33	.90/1.20	.40/.60	.20/.40	.20/.40	.08/.15
NE 9435	.33/.38	.90/1.20	.40/.60	.20/.40	.20/.40	.08/.15
NE 9437	.35/.40	.90/1.20	.40/.60	.20/.40	.20/.40	.08/.15
NE 9440	.38/.43	.90/1.20	.40/.60	.20/.40	.20/.40	.08/.15
NE 9442	.40/.45	1.00/1.30	.40/.60	.20/.40	.20/.40	.08/.15
NE 9445	.43/.48	1.00/1.30	.40/.60	.20/.40	.20/.40	.08/.15
NE 9450	.48/.53	1.20/1.50	.40/.60	.20/.40	.20/.40	.08/.15
NE 9537	.35/.40	1.20/1.50	.40/.60	.40/.60	.40/.60	.15/.25
NE 9540	.38/.43	1.20/1.50	.40/.60	.40/.60	.40/.60	.15/.25
NE 9542	.40/.45	1.20/1.50	.40/.60	.40/.60	.40/.60	.15/.25
NE 9550	.48/.53	1.20/1.50	.40/.60	.40/.60	.40/.60	.15/.25

CARBON-CHROME STEELS

	C.	Mn.	Si.	Cr.	Ni.	Mo.
NE 52100C	.95/1.10	.25/.45	.20/.35	.40/.60	.35 Mx.	.08 Mx.
NE 52100B	.95/1.10	.25/.45	.20/.35	.90/1.15	.35 Mx.	.08 Mx.
NE 52100A	.95/1.10	.25/.45	.20/.35	1.30/1.60	.35 Mx.	.08 Mx.

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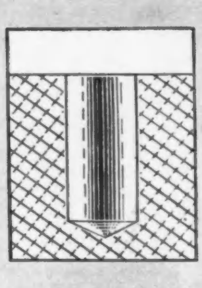
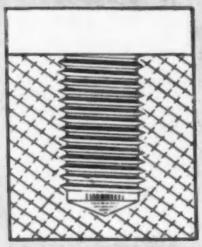
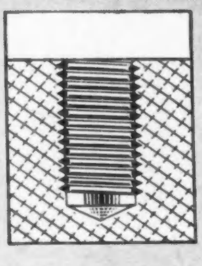


A Drill out old thread with drill having limits within plus .000 and minus .005 of diameter of Aero-Thread stud to be used. Depth of hole should be at least twice diameter of stud plus 1/8".

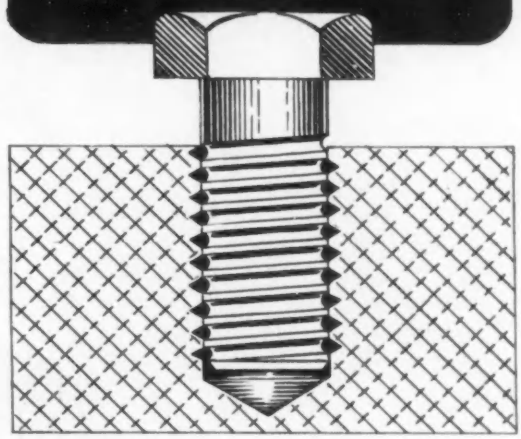


B Clean out and tap with Aero-Thread VAT Tap of proper size. Make sure that full depth of hole is tapped and that center line of tap is at right angles to face of boss when tapping.



C Place correct size insert over shank of inserting tool, with tang in slot of tool. Screw insert into tapped hole in clockwise direction. When insert is in position, shear off tang by reversing rotation of inserting tool about a half turn.

Replace with Aero-Thread* INSERTS



Aero-Thread Cap Screw in position in aluminum alloy boss. Note that thread profile of the screw has a circular valley and the tapped hole a modified 60° Vee thread form.

**Trade Mark Reg. U. S. Pat. Off.*

THE Aero-Thread Screw Thread System provides a thread lining for the tapped hole which also engages the threads of the mating screws. These inserts are made of hard, smooth, precision-shaped wire in stainless steel or phosphor bronze materials. Protection of tapped threads in light metals such as aluminum or magnesium, against wear and abrasion, is obtained at low cost with Aero-Thread Inserts. Holding power in light alloys and plastics is substantially increased and stripped threads eliminated.

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The Aero-Thread System is standard for replacement on combat aircraft engines, thus eliminating the oversize stud nuisance. It greatly simplifies field servicing. Expensive engine casings have been salvaged with the Aero-Thread System after they had been rejected for stripped threads. It is also used in original installations in aircraft engines, carburetors, oil filters, magnetos and other products. Send for Bulletins T-1A, No. 234 and Catalog No. 5.

U. S. Patent Nos. 2,150,875; 2,150,876; 2,152,681; 2,210,061; 2,244,824; 2,262,450; 2,257,089
British, Canadian and other foreign patents issued and pending.

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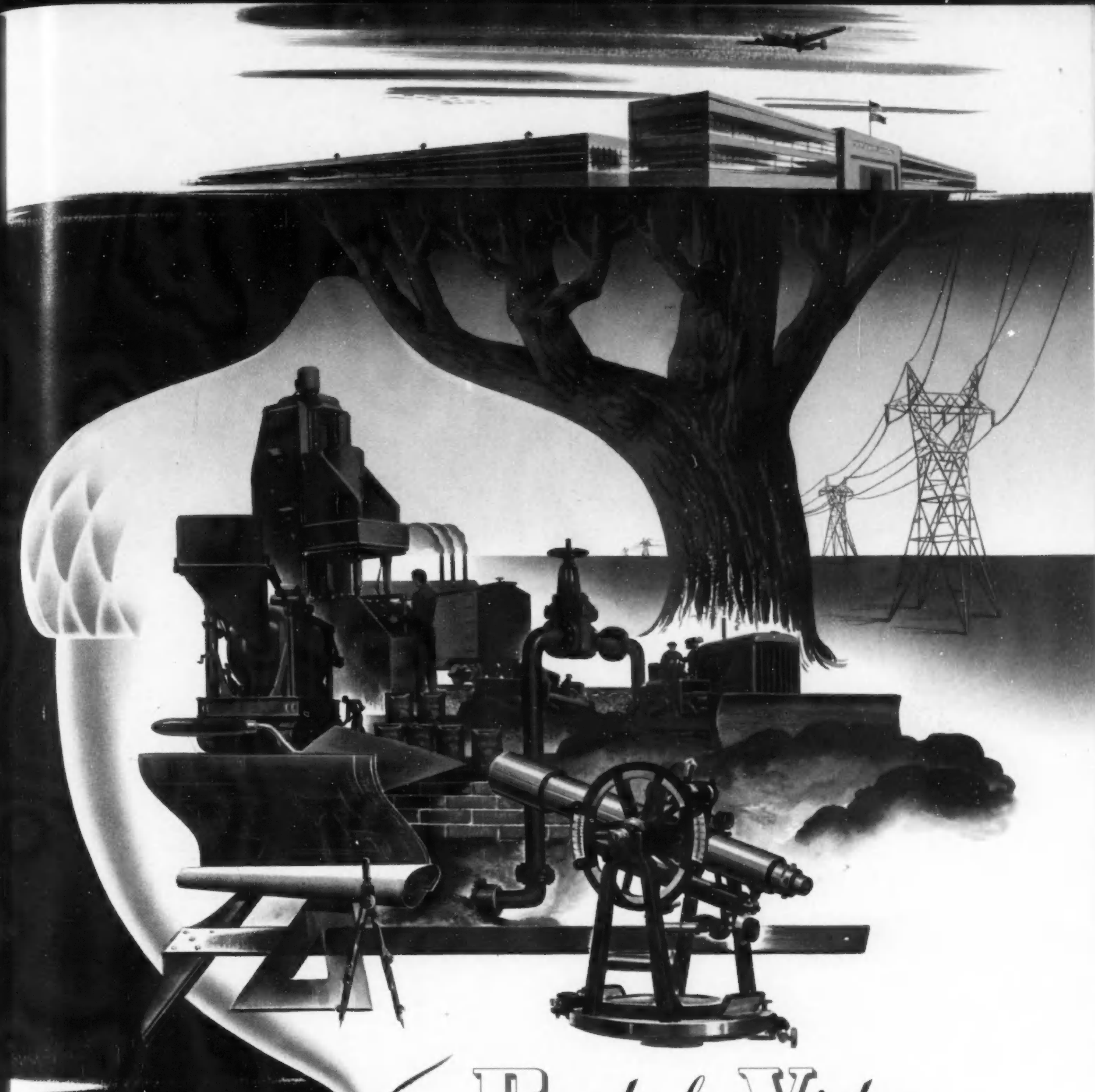

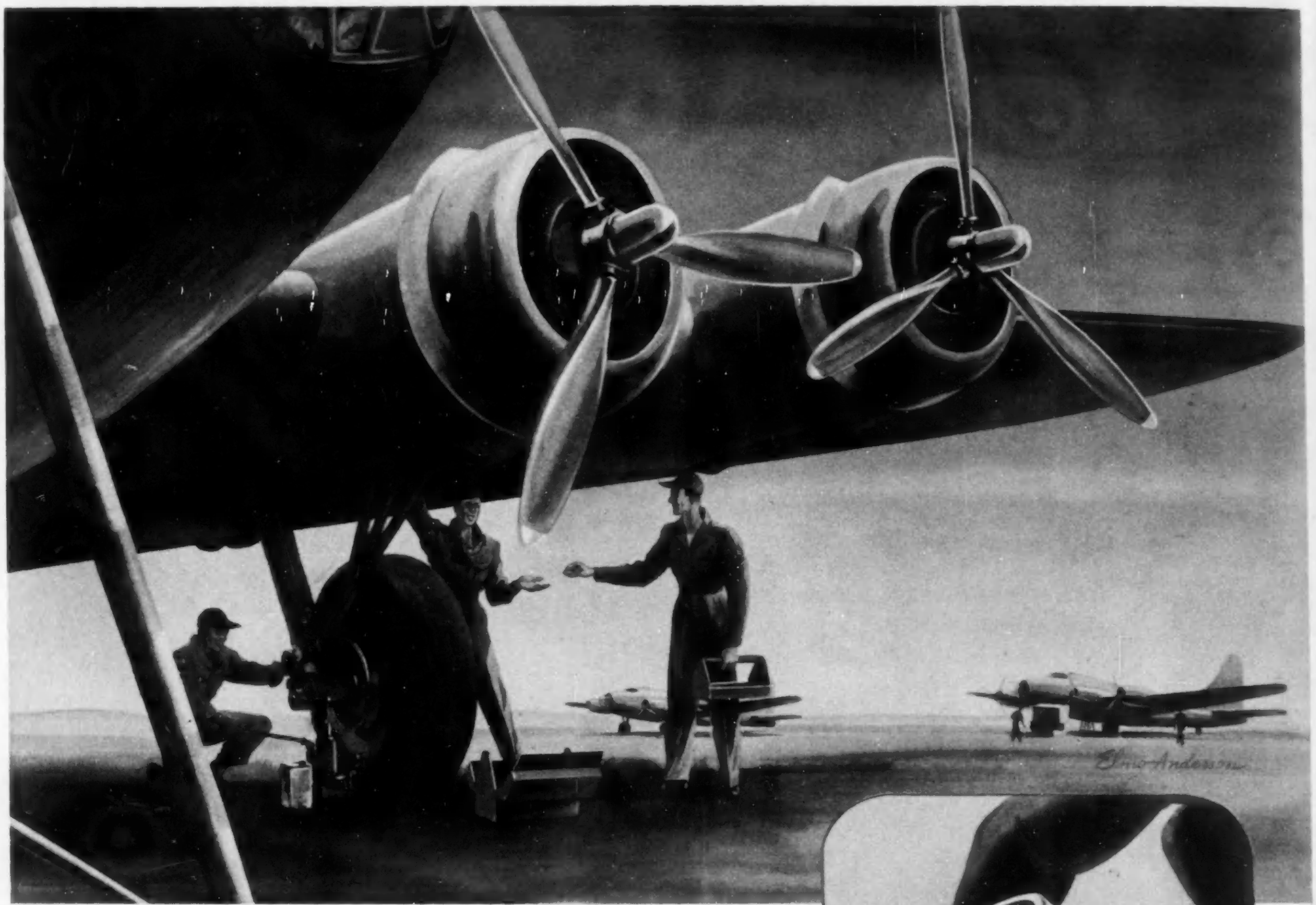

Field-Servicing a B-17E

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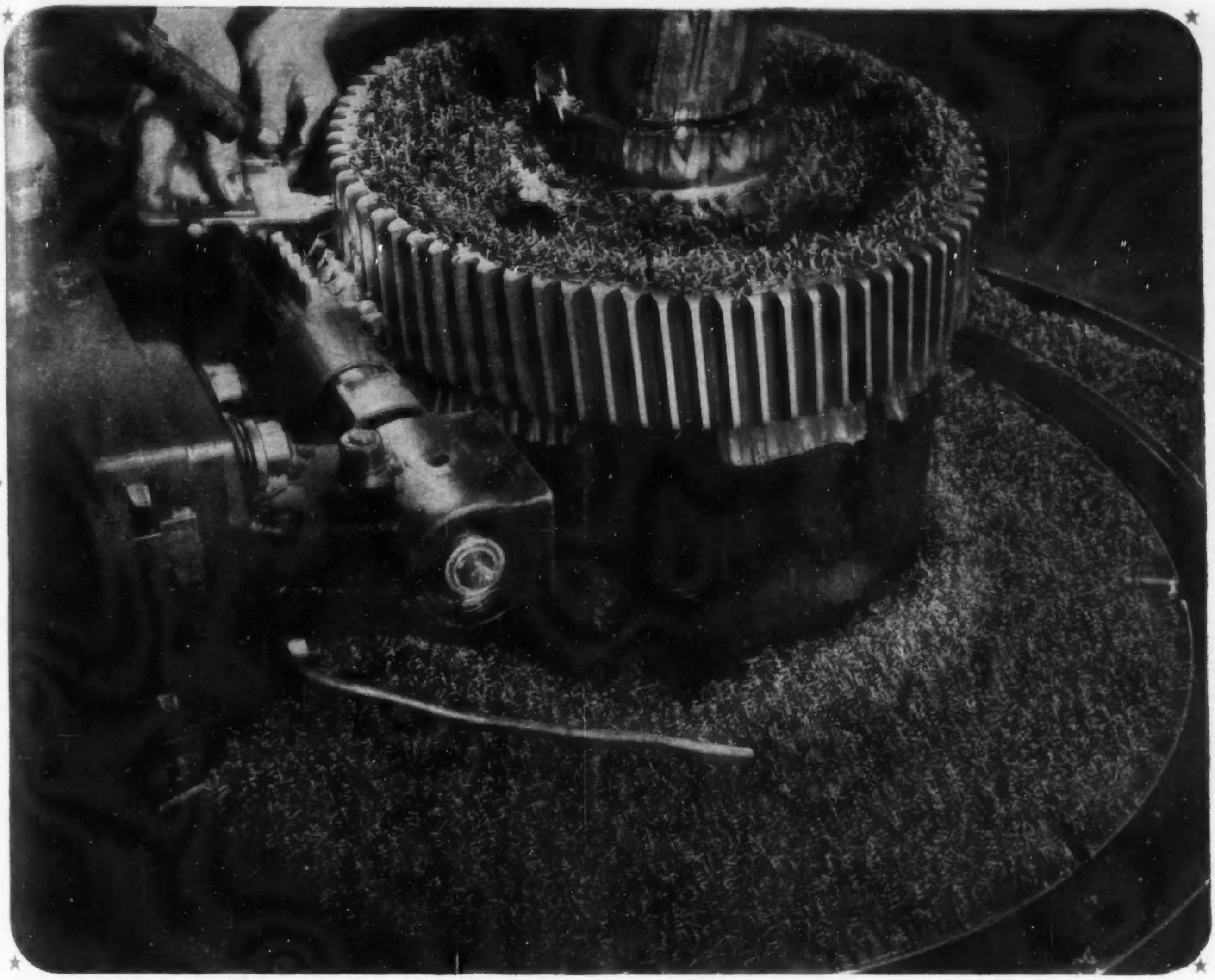

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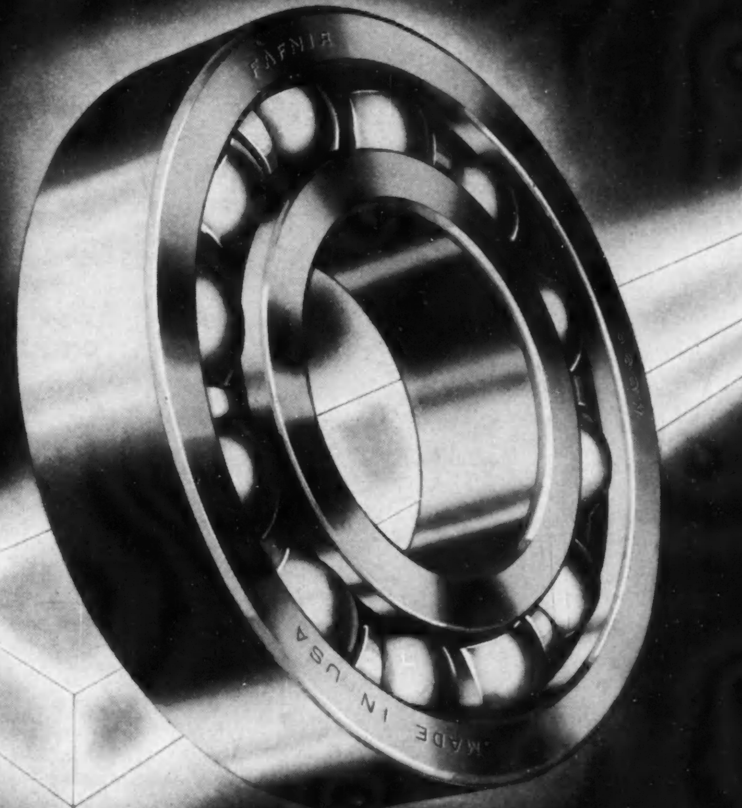
They are the Fafnir Ball Bearings that the Machinery Makers "bury"! They're big and rugged . . . built to last a lifetime in the *inaccessible spots.* They're the kind of ball bearings that require no pampering . . . and deep in the heart of a giant press, or blooming mill, or whatever . . . they work dependably and efficiently under the most punishing load and operating conditions. They're truly built to take a beating!

Fafnir Heavy Duty Ball Bearings, like all bearings of the complete Fafnir line, incorporate the time-proved Fafnir Balanced Design. This means larger balls and deeper races for greater strength and ruggedness . . . for greater radial and thrust capacity. Your requirements may not demand such brute strength in a ball bearing . . . but you are interested in the performance they typify which is reflected in all Fafnir Ball Bearings. The Fafnir Bearing Co., New Britain, Conn.

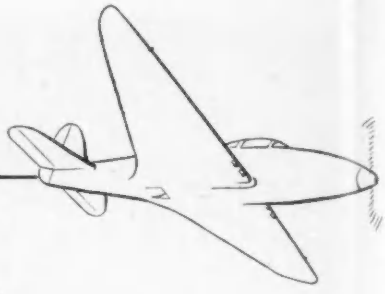


The Fafnir Ball Bearing pictured here is the No. 320 Medium Series, with a load capacity of 36,380 lbs. at 30 R.P.M. Designed for use on horizontal shafting to handle heavy radial loads. A corresponding bearing, for vertical shafting to handle heavy thrust loads, is the Fafnir Thrust Ball Bearing No. N1820.

ARMY E NAVY

FAFNIR
Ball Bearings

THE BALANCED LINE
FOR ORDNANCE, AIRCRAFT AND
INDUSTRIAL MACHINERY



Hard-boiled babies

Adolf...here's that bomber—*again.* This hard-boiled baby helped stall your *Wehrmacht* at Dunkerque...and just the other night it was back blasting your ports, hounding your shipping.

Hermann ... it's that Hudson your *Luftwaffe* has never really stopped...the ship the R. A. F. calls "Old Boomerang"—because it always comes back.

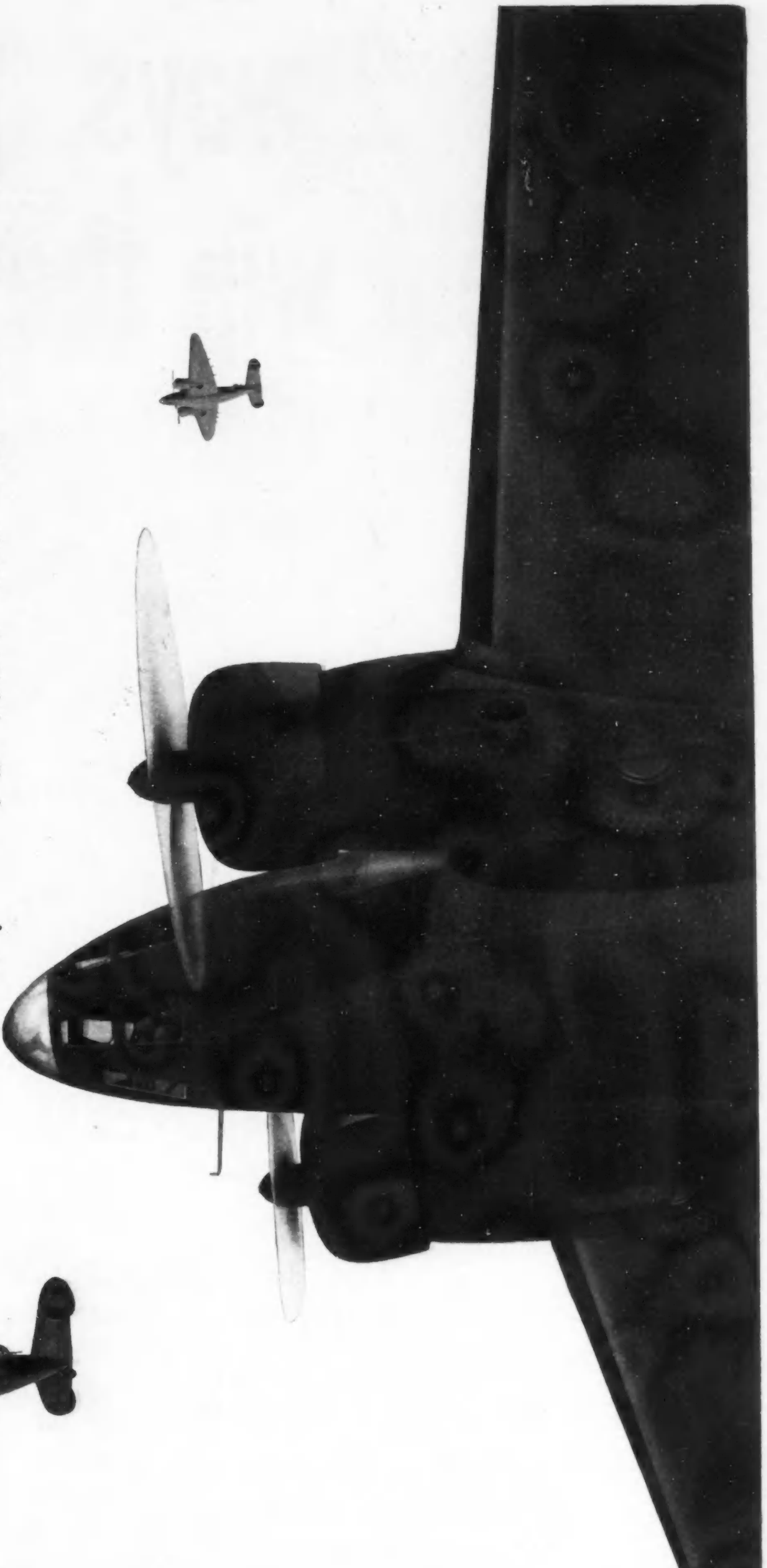
Hirohito...Tojo's met this Lockheed too, and he'll keep on meeting it ... lots of places, and doing lots of things. For the Hudson not only does a job; it invents new ones for itself!

And, boys, it's got a big brother, the Vega Ventura, that has all the family characteristics, and some new tricks of its own! Lockheed Aircraft Corporation ... Vega Aircraft Corporation ... Burbank, Calif.

for protection today, and progress tomorrow, look to

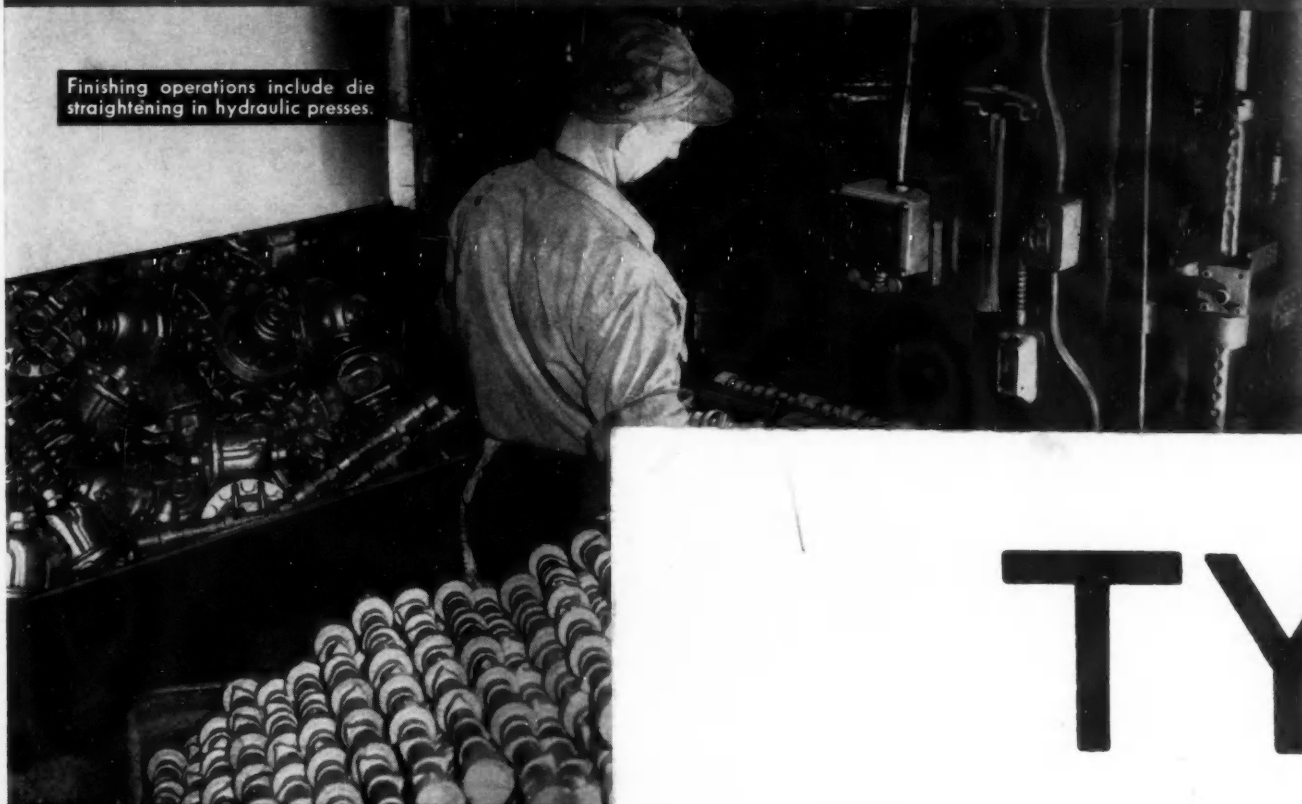
Lockheed

FOR LEADERSHIP



Member Aircraft War Production Council, Inc

CAST FOR A LEADING ROLE IN INDUSTRY



Finishing operations include die straightening in hydraulic presses.

TY

CUT-

Arm
Is Right Whe

There's no room for misfits in the daily sh
Steel castings that leave our foundry for v
in industry. Long before the finishing o
formed, any defective castings will ha
wayside under a barrage of checks and test
deep-etching, chemical analysis, electric-resistance tests, and others.

While flaws are few and far between, it is important that what rejections there are should take place in our plant, not yours. Your only concern is to take full advantage of the savings in machining time and tool wear that ArmaSteel castings effect in your manufacturing operations. ArmaSteel replaces critical materials and improves the performance characteristics of vital parts in a variety of applications. Write for information.

shafts, Hydra-Matic transmission gears, drive-shaft universal yokes, clutch throwout collars, valve rocker arms . . . Refrigerator crankshafts and connecting rods . . . Washing machine drive gears.

Saginaw Malleable Iron Division

General Motors Corporation Saginaw, Michigan

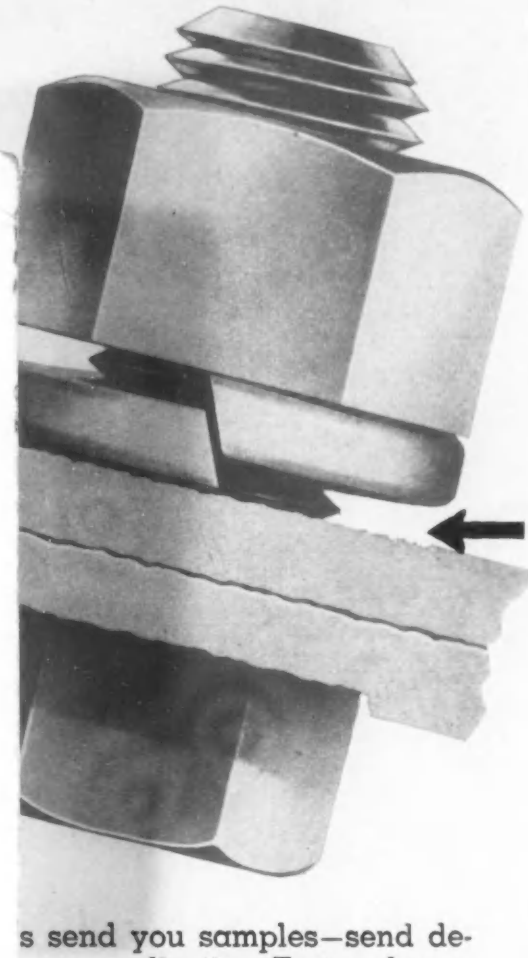
GM SAVE SERVE

*Reg. U.S. Pat. Off.

CAST FOR A LEADING ROLE IN INDUSTRY

A little spring can't go a long way

So when you order spring



assembly loosen sooner or later **unless** an adequate size helical **spring** washer is used to hold every part tight.

The parts will loosen because of bolt stretch, and frictional wear of metal on metal, burrs and flares, and because of pulverizing of paint, scale, and rust.

This is war time—be safer with powerful Kantlinks.

s send you samples—send de- your application. Test and compare them on the same job with **any** type of nut, or with any other type of washer. Kantlinks can't lose a real test. Try them for efficiency, economy and real safety.

Write today for descriptive folder.

THE NATIONAL LOCK WASHER COMPANY
NEWARK, N. J., U. S. A.

KANTLINK
TRADE MARK

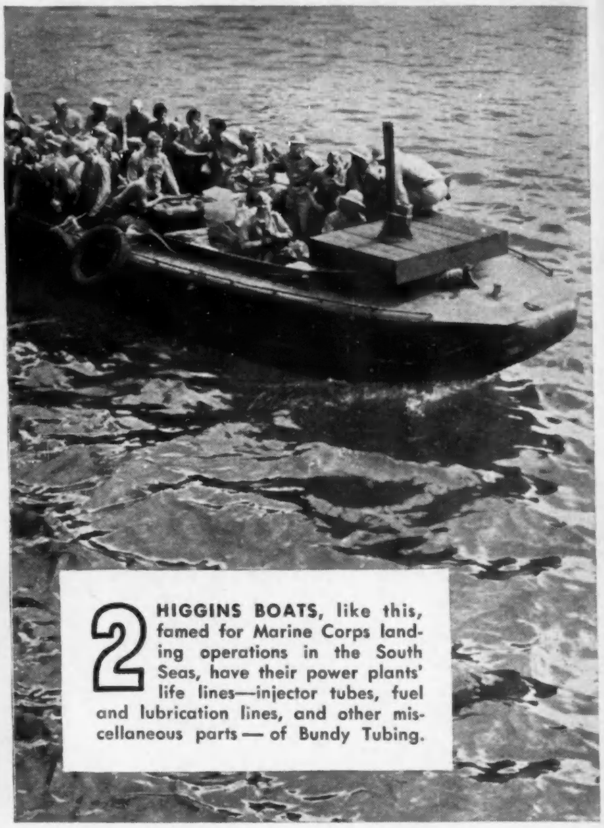
the long-range Spring Washer


Explosion Proof
BRAKE DRUMS
ELECTRIC FURNACE ALLOYED IRON
BRAKING SURFACE
STEEL SHROUD
Used on Bombers, Pursuit Ships, and Scout Cars
CENTRIFUGAL FUSING CO., Lansing, Mich.
Division of Campbell, Wyant and Cannon Foundry Co.

FAMOUS LIFE LINES

1 **AUSTRALIAN CONVOYS,** life lines of our South Pacific communications, are moving undisclosed quantities of war material and reinforcements to our troops and allies down under. Here is a gun crew on a transport at its daily job of testing its anti-aircraft protection.

Official U. S. Navy Photograph



2 **HIGGINS BOATS,** like this, famed for Marine Corps landing operations in the South Seas, have their power plants' life lines—injector tubes, fuel and lubrication lines, and other miscellaneous parts — of Bundy Tubing.

Official U. S. Marine Corps Photograph

RIP CORD GRIPS

Rip cord grips for parachutes, life lines of safety for American airmen, are one of the important new wartime uses which are taking thousands of feet of Bundy Tubing.

EXPANDED Bundy plants, working round the clock, are turning out hitherto undreamed-of quantities of Bundy Tubing — in an ever-growing variety of types and sizes.

Where does all this Bundy Tubing go? Here are a few of thousands of war uses:

A battleship (or for that matter any other type of war or cargo vessel) has innumerable Bundy Tubing parts in its cold rooms for food and munitions, in cooling systems and individual refrigerators, in auxiliary power plants, in surgical lamps and radios.

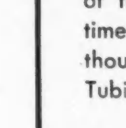
The army — whether in training camp or in the field—uses Bundy Tubing for the applications previously mentioned—plus Bundy parts in automotive and tank equipment, in planes and gliders, for parachute grips and component parts of ammunition.

Industry in peacetime had long accepted Bundy Tubing as standard for many purposes. It is natural that this preference should carry over into similar war applications and into an ever-increasing number of new uses. Bundy Tubing Company, Detroit, Michigan.

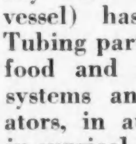
BUNDY TUBING

Buy U. S. War Bonds — *Get In Your Scrap*

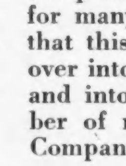
ENGINEERED TO YOUR EXPECTATIONS



BUNDYWELD double-walled steel tubing, hydrogen-brazed, copper-coated inside and outside. From Capillary sizes up to and including 1/16" O. D. This double-walled type is also available in steel, tin-coated on the outside, and in Monel.



BUNDY ELECTRICWELD steel tubing. Single-walled — butt welded — annealed. Available in sizes up to and including 2" O. D. Can be furnished tin-coated outside in smaller sizes.



BUNDY "TRIPLE-PURPOSE" tubing. Double-walled, rolled from two strips, joints opposite, welded into a solid wall. Available in all Monel; all steel; Monel inside—steel outside; Monel outside—steel inside. Sizes up to and including 5/8" O. D.

30 Ounce Miracle
that Revolutionizes Starting Ignition

"MIRACLE?" Well, miraculous at least for the way it is strengthening American air might in the frozen wastes that may become the main thoroughfare of this global war.

For as aviation engines have constantly grown in size and power, one major difficulty has been the problem of providing adequate all-climate starting ignition.

Today this problem has been solved by the American Bosch Induction Vibrator . . . a simple device small enough to put in one's pocket. From the Equator to the Poles—winter or summer—engines in United Nations planes can now start in a flash and get into the air more quickly.

This development—begun in times of peace under the American Bosch policy of relentless research in every phase of its field—has been completed in time to meet intensified military demand. Broadening the potential battle zone of every plane in which it is installed, it serves the cause of Victory by helping to open new fronts for American and Allied air strategy.

★ ★ ★

American Bosch Corporation • Springfield, Mass.

Craftsmen in the New England Tradition

★ AMERICAN BOSCH ★

AVIATION & AUTOMOTIVE ELECTRICAL PRODUCTS . . . FUEL INJECTION EQUIPMENT

SEGREGATED Scrap... is your link to VICTORY!

Scrap salvage is a vital factor in the war effort. To be fully effective a scrap program must include methods for segregating and conserving critical alloying elements so urgently needed in the construction of tanks, guns, ships and planes.

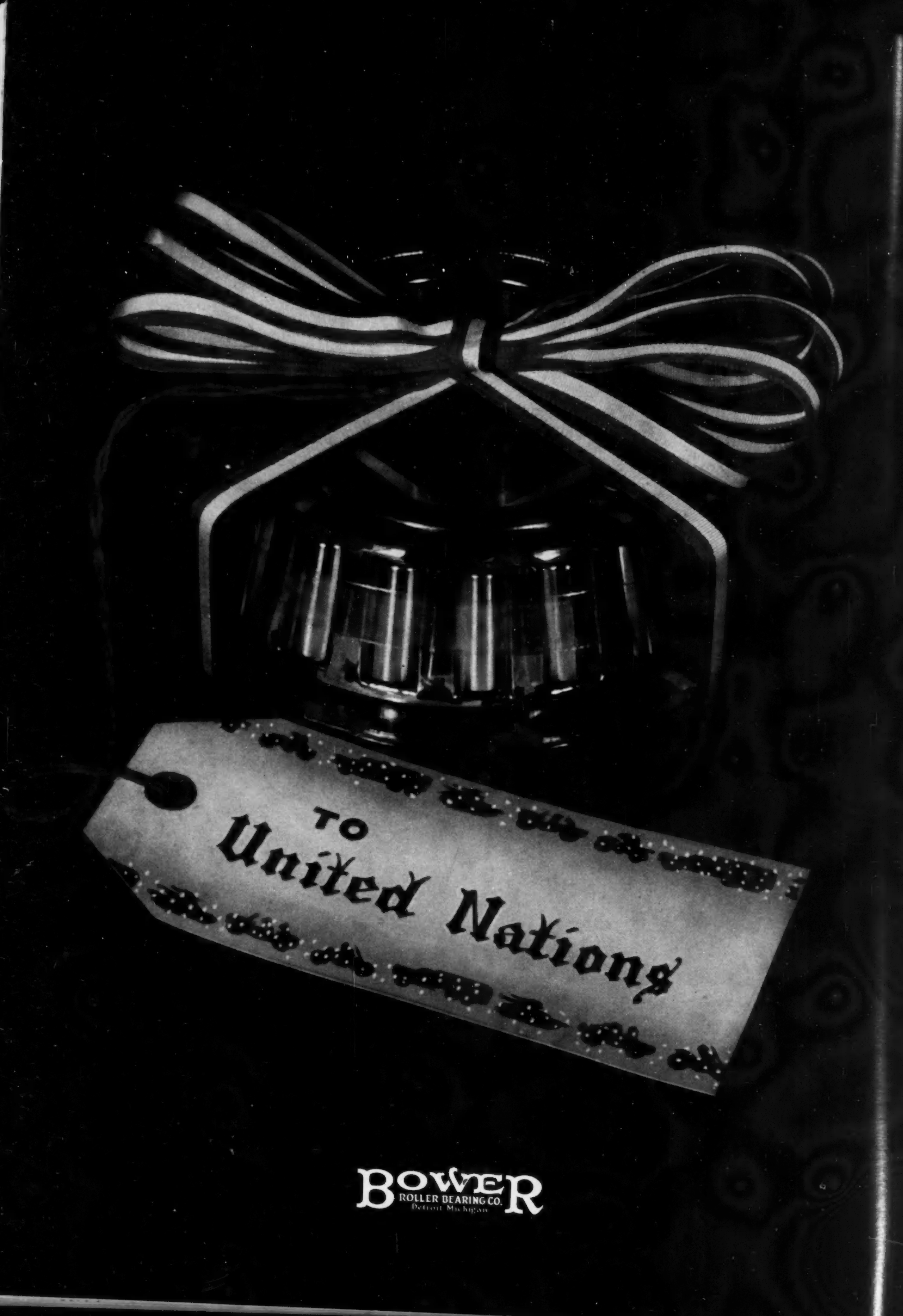
Ferrous and non-ferrous metal scrap should be collected in separate containers at the machine where they are generated. Each class of high-speed tool steel and each type of constructional alloy steel should likewise be kept separate so that the alloy content can be returned to service.

Remember, — alloy scrap which is segregated, classified and labeled according to type and composition is a vitally important commodity today—and urgently needed to augment primary supplies of Nickel, molybdenum, tungsten, etc.

The metallurgical experience of our technical staff is available to aid you in these and other phases of metal salvage.

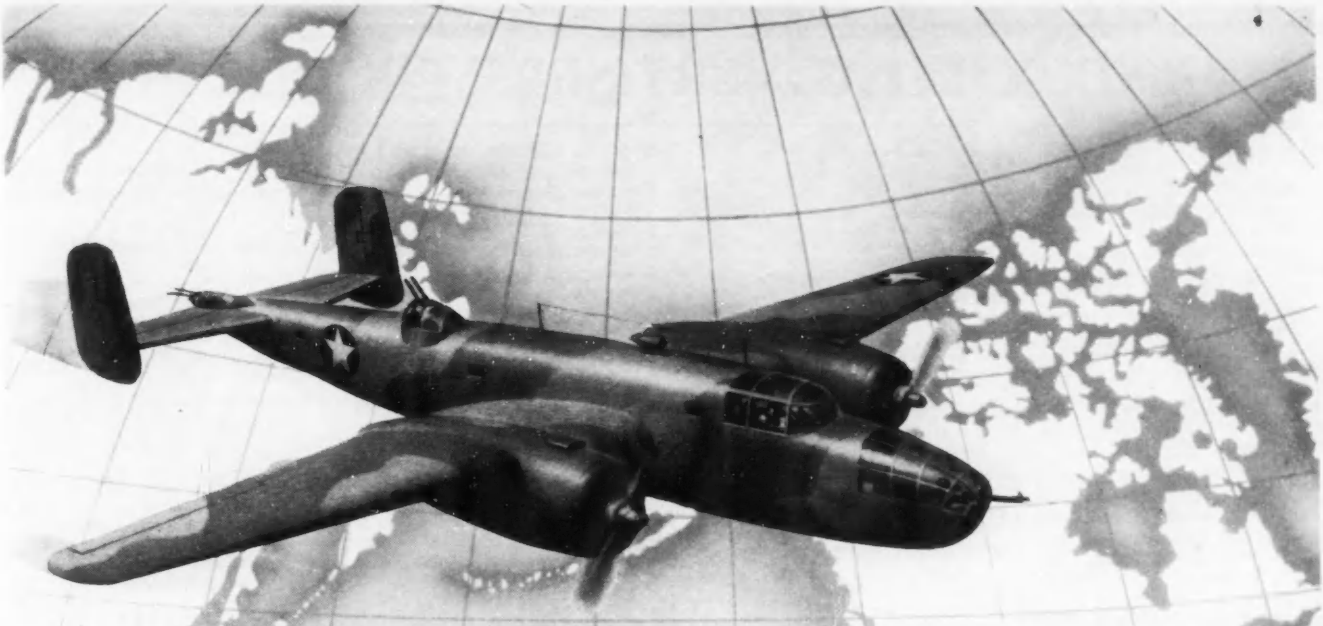
KEEP SCRAP MOVING INTO WAR PRODUCTION!

THE INTERNATIONAL NICKEL COMPANY, INC. 67 WALL STREET NEW YORK, N. Y.


TO
United Nations
BOWER
ROLLER BEARING CO.
Detroit Michigan


Almighty God, We Make Our Earnest Prayer . . .
"Almighty God, we make our earnest prayer that Thou will keep the United States in Thy holy protection; that Thou will incline the hearts of the citizens to cultivate a spirit of subordination and obedience to government; to entertain a brotherly affection and love for one another and for their fellow-citizens of the United States at large." GEORGE WASHINGTON
Spicer Manufacturing Corporation • Toledo, Ohio
39 YEARS OF
Spicer
SERVICE
BROWN-LIPE
CLUTCHES and TRANSMISSIONS
SALISBURY
FRONT and REAR AXLES
SPICER
UNIVERSAL JOINTS
PARISH
FRAMES STAMPINGS


PUT JOBS LIKE THESE
ON A MULT-AU-MATIC
IF YOU REALLY NEED
FAST PRODUCTION
THE BULLARD COMPANY
BRIDGEPORT, CONNECTICUT



INDEX TO ADVERTISERS' PRODUCTS

(Continued on page 86)

INDEX TO ADVERTISERS' PRODUCTS

(Continued from page 84)

(Concluded on page 88)

INDEX TO ADVERTISERS' PRODUCTS

(Concluded from page 86)

The addresses of companies listed in this index can be obtained from their current advertisements indexed on page 90.

■ INDEX TO ADVERTISERS ■

The Index to Advertisers' Products is given on pages 84, 86 and 88